DETAILS of MACHINERY

F. CAMPIN, C.E.

DETAILS OF MACHINERY

COMPRISING

INSTRUCTIONS FOR THE EXECUTION OF VARIOUS WORKS IN IRON

IN THE

FITTING-SHOP, FOUNDRY, & BOILER-YARD

ARRANGED EXPRESSLY

For the use of Draughtsmen, Students, and Foremen Engineers

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Capio Lumen

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PREFACE.

In introducing the present Treatise to the Engineering public, there is but little to be said by way of Preface. It is believed that hitherto no work has been specially devoted to *details* of machinery, and having noticed the want of such an one amongst Draughtsmen and Foremen Engineers, the Author has been induced to prepare the present volume.

The Author has throughout adhered strictly to simple arithmetic, not having used even a *plus* or *minus* sign in any of the calculations, which are illustrated by examples worked out in full, in order to render the mode of working the rules, practically, perfectly clear.

The results generally are only carried out to two or three places of decimals, which is amply sufficient for all practical purposes.

Leeds, *November*, 1882.

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DETAILS OF MACHINERY.

INTRODUCTION.

In designing any kind of structure, or machine, there may be considered two operations, almost distinct from each other, to be performed. The first consists in determining the general principles upon which such work depends for its due fulfilment of the purpose for which it is proposed, and the second, the exact arrangement of the details, so as to combine them according to the general plan adopted. The projector of the work will arrange, of course, the general design, but it very often becomes the duty of the manufacturer to arrange the constructive details; and, in order to obtain a creditable result, it is frequently necessary to devote much care and consideration to these minor points.

In the present treatise it is purposed to consider especially the form and arrangement of certain classes of machinery and elements of structures, which are in general demand; and in so doing it must be distinctly understood that the *general principles* of structures and machinery will not be alluded to, except in such cases as it may seem necessary so to do in order to elucidate the remarks relating to details properly under consideration. Not only is it necessary to estimate the amount of strain on the

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various details of machinery, but in most cases as much or even more consideration is necessary to ascertain exactly the way in which the strain acts, which is not in all instances very obvious at first sight; in fact there are many important points now mooted, upon which decided differences of opinion exist, so that we find two engineers, each entitled to equal respect, holding opposite views on the same point. Nor is this state of affairs very difficult to account for, for it will subsequently be seen that a theory may apply very well in one shop, but that it will not obtain in another, where there exist different methods of manufacture ; hence, in each case, we have to regard not only the nature of the work to be done by any element, but also the manner in which such element will be constructed. It is an ancient truism that there is a right and a wrong way to do everything; but this does not go far enough, at least in mechanical engineering, for there may be two or three right ways of doing a job, and they may in fact be all equally right, although they are preferred differently by different individuals; hence we should be prepared to understand all the ways in which any portion of work can be done to attain the desired end, after which we can select that method which, according to our judgment, appears best qualified to satisfy our requirements.

We may class the strains to which machines and structures are liable under the following heads: pressure without motion, or static strain, and pressure with motion, or dynamic strain. The first is simple to deal with, as its amount is usually known and is not variable. If the strain be in tension, it is a constant force endeavouring to tear the element asunder; if in compression, the element acts as a pillar ; or if it be a cross strain, the element acts as a *beam*. *But in the* second class of strains the intensity of

the force is constantly varying in many instances, and often tends to set up and maintain a series of vibrations intermittent or continuous; and if continuous, probably increasing in range and in violence so long as the machine continues in motion. Where the force acting on an element impresses it at intervals with a tendency to deflect, and at each impulse in the same direction, a vibratory motion will be set up, which will increase until the elastic resistance of the material opposes such resistance that the successive impulses, although sufficient to maintain the vibration, are not powerful enough further to increase it; here then the material must be disposed in such a form as is best suited to resist the force, according to the particular direction in which it will act. In another case we may observe forces causing vibratory motion but acting in opposite directions alternately, hence the material must then be distributed to withstand the impulse equally well in both directions; and in some instances of revolving elements they must be adapted to withstand force equally well in all directions.

It will be readily seen that, the actions of strains on mechanical elements being so varied, it is necessary to be acquainted with the properties of different materials in order to determine of what kind any particular detail should be made; thus, where blows and violent concussions are anticipated, some material must be used which, yielding by its elasticity, gradually takes up the strain, instead of opposing it instantly. It is easy to show how the property of yielding *actually reduces* the maximum force to which the element is subjected; thus, let us suppose that we have a weight of 50 lbs. falling through a distance or height of 10 feet; at the end of its fall there will be accumulated in it work equal to 50 lbs. raised 10 feet, or 500 *foot lbs*. Suffer this accumulated work to be

expended on a bar of metal, and let there be two bars of different materials and two equal weights falling upon them, then upon each bar there will be expended 500 foot lbs. of work, which will be done while the weight is passing through a distance equal to the amount of bending or deflection of the bars corresponding to the force of the blow.

The bars being of different materials will also exhibit different degrees of elasticity; let it, therefore, be supposed that one bar bends through one inch under the blow, and that the other one deflects through two inches under the same blow; knowing the amount of accumulated work expended, we can find the mean force by dividing that accumulated work by the space in which it is expended. We have 500 foot lbs. expended in passing through one inch, or one twelfth of a foot; hence, dividing by one twelfth, we find the mean pressure to be 6,000 lbs. pressure on the bar during the blow; in the second case we have to divide by two inches, or one sixth of a foot, so we find in this case the mean pressure is only 3,000 lbs. on the bar. Hence it is evident that the strain produced by a blow upon a mechanical element (the blow being constant) will vary universally as the range of elasticity of the material, so that the material best adapted for static load will often be unsuitable altogether to be opposed to concussion.

We will now briefly refer to the properties of materials most generally used in the construction of machinery, &c.

It has been shown that the properties of materials, considered practically, vary principally in two respects: strength to resist fracture under steadily imposed forces, and elasticity, or the property of yielding, to some extent, by change of external form under the influence of external pressure; and *such yielding does* in all cases occur to a greater or smaller

degree previous to rupture, supposing that the strain is carried to the breaking point. The ultimate strength of materials is at once expressed in figures: thus we know that good iron bars will break at a tensile strain of about 22 to 24 tons per square inch of sectional area, though some of very superior quality, such as Farnley iron, will sustain a strain of 30 tons per square inch before breaking. Cast iron will break in tension at 7 tons per sectional square inch; but in resisting compression it will sustain from 45 to 50 tons per square inch. Thus much for actual resistance to present rupture; but of elasticity there is more to be said, as by understanding the proportions in which it exists we are enabled to judge what strains are likely to produce internal injury to materials, injury which, though unaccompanied by external symptoms, may be the commencement of a failure in strength which, being continued by subsequent strains, ultimately leads to the rupture of the element under a strain far beneath its ultimate strength in the first instance.

The differences of elastic properties in various bodies may without difficulty be traced to differences in chemical composition and molecular arrangement, but it is not our intention to enter upon these points in the present treatise, as their elucidation would not assist the more especial researches to which our efforts are directed. The elasticity of solids is exhibited under two distinctive phases, "*range of elasticity*," and "*perfection of elasticity*." Indiarubber has a great range of elasticity, and glass has a very perfect elasticity, with, however, a comparatively small range; the range is shown by the extent to which a body may, in respect to its size, be distorted without rupture, and the perfection by the nearness to its original form and size which it will *resume* after being distorted for long periods.

A band of indiarubber, we know by common observation, soon becomes permanently stretched by continued extension; but a strip of good glass may be bent slightly for years, and will then resume its original shape and size when the deflecting force is removed.

We have above explained that, in some cases, range of elasticity is preferable to perfection of elasticity, as to resist concussion; thus, though cast iron has more perfect elasticity than wrought iron, the latter is best adapted to withstand a blow, in the same way that glass is *brittle* and indiarubber is not.

So long as any strain, put upon a machine or any portion of a machine, does not impair the *perfectness* of its elasticity, the ultimate strength of the material remains intact, but directly the material is *permanently* altered in form or dimensions, its strength is actually reduced, so long as that material is in the solid state. It may, therefore, be concluded that no material should be subjected at any time to a strain so great that when such strain is removed it fails to recover its previous form and dimensions, or, in technical terms, it should not be strained so far that it will take a "permanent set," for until that point is reached the elasticity (or power of recovery from strains or blows) remains intact; it may also be noticed as a matter of common prudence that this point should not be nearly approached, but a fair margin allowed for emergencies; thus good bar iron will not take a permanent set under about ten tons tensile strain per sectional square inch, but it is not usual in practice to load it with a strain exceeding five tons per square inch of sectional area.

CHAPTER I.

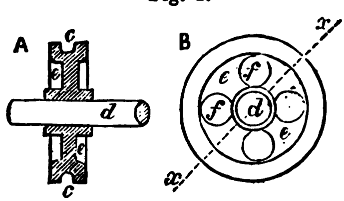
PULLEYS, CHAINS, AND HOOKS.

PULLEYS, Chains, and Hooks, may be conveniently treated under one head, as, although they are not identical in principle, they are usually associated together in one machine, as in a crane or winch.

Pulleys, if not of large diameter, are almost invariably made solid in the centre, that is to say, without arms, though sometimes the disc or central part is lightened out by holes being cast in it.

In Figure 1, A and B respectively represent a vertical section and a side elevation of any ordinary cast iron pulley or snatch block, the section A being taken on the line *x - - - x*. *c c* shows the groove in the pulley to receive the rope or chain it is intended to carry or to work on; *d* is the gudgeon or centre, which in small pulleys is usually fixed so that the pulley revolves freely upon it, but in large pulleys the gudgeon is firmly keyed on and revolves in bearings provided to receive its ends; *e e* shows the disc or central part of the pulley, which may or may not be perforated as shown by the circular holes *f f*.

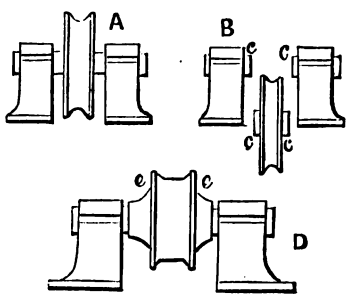
Fig. 1.



In very *small* pulleys the disc e e is kept as thick as

the periphery measured over the groove *c*, but for larger ones, as this amount of metal is not necessary, it may be thinned down as shown, but care must be taken to leave sufficient metal at the part *g* to allow for the groove wearing deeper in working. The requisite thickness of the disc will presently be shown. In all cases where the load on a pulley is sufficiently important to require calculation of the strength of the parts, the centre or gudgeon upon which it revolves should be supported at both ends, in order that there may be no bending strain upon it, and that in fact the strain should be shearing force, which it will be if the bearings be put sufficiently close to the centre boss of the pulley, as shown at A Fig. 2, where rupture would occur, as shown at B, the gudgeon having been sheared through between the edges of the pulley boss and the bearings, as shown at *c c c c*, one portion of the gudgeon being left in the pulley, the two ends remaining in the bearings. It may occur,

Fig. 2.



however, that the bearings cannot be put near enough to be close to the pulley if made of the usual thickness; then the boss may be swelled out as shown at *e e* in the view D, so as to spread the bearing of the pulley over the whole length of the gudgeon, between the two fixed bearings. Having determined the nature of the strain upon the gudgeon or centre, its size may be determined according to the load the pulley is designed to carry. Supposing the gudgeon to be, as it usually is, made of wrought iron, we

may allow, for the safe working strain upon it, four tons per square inch of sectional area; and in case of rupture it will have to be sheared through in two places as shown at B. Let the pulley be designed to sustain a load of sixteen tons in all, then there will on each section be a shearing strain of eight tons. As there are two sections carrying the load, let us see what diameter of the gudgeon is requisite. Allowing four tons safe load per square inch, it will require two square inches to sustain the weight, hence the sectional area of the gudgeon must no where be less than two square inches. The required area of the gudgeon being known, to find its diameter we must divide the said area by 0·785 and extract the square root of the quotient: thus—

```
·785)2·000(2·5477, &c.
     1570
     ----
      4300
      3925
      ----
       3750
       3140
       ----
        6100
        5495
        ----
         6050
```

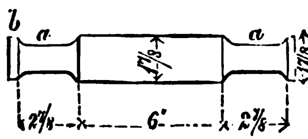
Taking the square root we have,

```
   1)2·5477(1·59, &c. inches.
     1
     ---
  25)154
     125
     ---
 309)2977
     2781
     ----
      196
```

or say $1\frac{3}{8}$ inch in diameter at the smallest part, which would be in the journals where the gudgeons rest upon the bearings.

The gudgeon might therefore be of the form and size shown in Fig. 3, assuming the distance between the plummer blocks, in the clear, to be six inches. The journals *a a* are made $1\frac{3}{8}$ inch in diameter. The length for a wrought iron journal should be about one and three quarter times the diameter, hence in this case it would be $2\frac{3}{8}$—$\frac{1}{32}$ but $2\frac{3}{8}$ inches is sufficiently near. The journals may have collars *b* on the end or not. In such a case as this there is no thrust in the direction of the length of the gudgeon, hence the collars may be small; they are shown $\frac{1}{8}$ of an inch deep, so that the body is $1\frac{7}{8}$ inch in diameter.

Fig. 3.



Having thus determined the proportions of the gudgeons, we can now pass on to those of the pulley:—

The diameter of the boss of the pulley should be twice that of the gudgeon, where it passes through the pulley; and if it be made with arms, each arm must be made strong enough to sustain the whole weight upon the pulley, for it may happen that with four-armed pulleys the whole weight is actually upon one arm. In the case taken above, the load is sixteen tons. The nature of the strain is compressive, and taking the crushing strength of cast iron at 50 tons, we may for machinery assume one tenth as the safe working load; hence that load should not exceed five tons; then dividing 16 tons by 5 we find the area of each arm of the wheel—

$$5)\underline{16{\cdot}0}$$
$$3{\cdot}2$$

If the pulley have four arms, each one may be made of $+$ section measuring 3 inches wide and $2\frac{1}{4}$ inches in breadth, the metal being $\frac{3}{4}$ of an inch in thickness. The sectional area is thus found: that of the broad or central web will be 3 inches by $\frac{3}{4}$, which is equal to (3 multiplied by 3 and divided by 4) 2·25 square inches; the feathers on each side of the web will be $\frac{3}{4}$ inch by $\frac{3}{4}$ inch thick, for the two $\frac{3}{4}$ feathers added to $\frac{3}{4}$ thickness of web will make up the $2\frac{1}{4}$ inches, and two $\frac{3}{4}$ inch feathers, $\frac{3}{4}$ inch thick, have together an area of 1·125 square inches, which added to 2·25, the area of the web, equals 3·375 square inches, which is somewhat in excess of the required area.

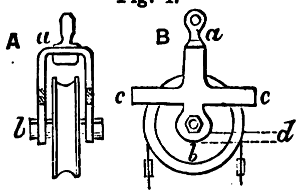
If the pulley have six arms, one arm will at no time carry more than half the total load; hence there is a saving in having six arms, inasmuch as, while the number of arms is only increased from 4 to 6, the area and weight of each arm is reduced to one half.

When the central parts of the pulleys are made solid, the thickness should, according to theory, decrease towards the periphery, but as such wheels are usually of small diameter this is not generally so arranged. In order to determine the thickness of the disc at the boss, we may consider the load carried by an arm having a breadth equal to the diameter of the boss, which in the above case would be $3\frac{3}{4}$ inches; hence, if we made the disc 1 inch thick, there would be sufficient strength, for that would give 3·75 square inches, being somewhat over the 3·2 square inches required by calculation. In very large pulleys the size of the arms becomes rather a matter of proportion, when the loads are light. If the pulley gudgeon runs in ordinary plummer blocks there is no need to calculate the metal required in them, as there is always plenty and to spare; but it may be mentioned that where brass bearings are used it is unneces-

sary to have two brasses to each journal, as the strain always being in one direction the wear is also all on one side of the centre, which side, therefore, alone requires a brass.

The greater number of small pulleys used as snatch-blocks are carried in bearings of the form shown in Figure 4, and these have fixed bolts upon which they revolve freely.

Fig. 4.



In Fig. 4, A and B show two elevations of an ordinary snatch-block, as used for cranes, &c.; the pin upon which runs the pulley or "sheave," as it is often termed, being carried by the forked part *a b*; *c c* are "wings," the object of which is to prevent the chain from slipping out of the groove in the pulley, in case of the latter being accidentally tipped over from its normal position; *d* shows the distance from the outside of the bolt or pin to the end of the forked bearing *a b*. In this case the tendency to rupture lies in the direction from the bolt or pulley centre towards *b*, and in case of failure it will occur by the direct splitting open of that part. In the first place it is necessary to determine the requisite width and thickness of the upper parts of the forked bearing or strap *a b*, which, of course, has to carry the same amount of strain as is on the pulley. Each side should have a sectional area equal to that of the bolt carrying the pulley, or it may be determined from the strain direct. Assume the width to be twice the diameter of the bolt, then the required thickness will be found from the following rule:—

RULE.—*To find the thickness of one side of the suspending jaw in inches, multiply the diameter of the bolt in inches by 0·39.*

Let the diameter of the bolt be $1\frac{1}{4}$ (1·25) inches, then the breadth of the suspending jaw being $2\frac{1}{2}$ inches, the thickness of each side should be thus found—

1·25 inches diameter of bolt

0·39

1125

375

·4875 inches.

So that we shall have ample strength by making the thickness half an inch. If a different ratio of width be adopted, the following general rule should be worked to.

Rule.—*To find the thickness of one side of the suspending jaw in inches, divide the square of the diameter of the bolt in inches by the width of the jaw in inches, and multiply the quotient by* 0·78.

Let the diameter of bolt be $1\frac{1}{4}$ inches, width of jaw 2 inches, then the thickness will be thus found:—

1·25 inches diameter of bolt

1·25

625

250

125

Width of jaw 2)1·5625 square of diam. of bolt

·7812

·78 multiplier

62496

54684

·609336 inches thickness of jaw.

This thickness would in practice be made $\frac{5}{8}$ inch.

We may here observe that it is not usual either to square or cube numbers in actual working, nor to extract the square or cube roots, but to take such processes from

tables in order to save time; and having shown the methods of working these two processes; we shall in the remaining part of this treatise dispense with the working in full, which occupies much space which we require for other purposes. The tables used will be those contained in the author's "Engineer's Pocket Remembrancer," though of course any other accurate tables of squares and cubes, and square and cube roots, will serve equally well.

The thickness of one side of the jaw being determined, it remains to ascertain the length, marked *d*, which may be obtained from the following rule:—

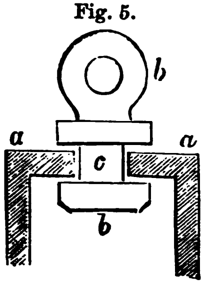
RULE.—*To ascertain the length of the jaw beyond the bolt in inches, divide the square of the diameter of the bolt in inches by the thickness of the jaw in inches.*

Let the diameter of the bolt be $1\frac{1}{4}$ inches, and the thickness of the jaw $\frac{1}{2}$ inch. The square of $1\frac{1}{4}$ inches is 1·56 inch nearly. This is to be divided by $\frac{1}{2}$ inch, which is equivalent to multiplying it by 2. Hence we have—

$$\begin{array}{r} 1{\cdot}56 \\ 2 \\ \hline 3{\cdot}12 \text{ inches} \end{array}$$

In the next place must be considered the arrangement by which the jaw itself is supported. It is shown in Fig. 5.

Fig. 5.



The upper part of the jaw is shown in section at *a a*, and through it passes a stud *b b*, formed with an eye at the top for attachment to a chain or other means of support. The neck of the stud, which is that part shown at *c*, should not be less in area than twice the area of the bolt upon which the pulley is carried; and to obtain this area the *diameter of c must* be equal to 1·414

times the diameter of the bolt carrying the pulley. Practically, we may call this 1·5 ; hence we have the following rule for the diameter of the smallest part of the stud *b b′*:—

Rule.—*Make the diameter of the stud* (b b′) *equal to the diameter of the bolt carrying the pulley, multiplied by* 1·5.

In the previous case the least diameter of the bolt to carry the pulley is 1¼, or 1·25 inches; hence the least diameter of the stud should be found thus :—

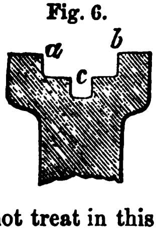
$$\begin{array}{r l} 1{\cdot}25 & \text{inches diameter of bolt} \\ 1{\cdot}5 & \\ \hline 625 & \\ 125 & \\ \hline 1{\cdot}875 & \end{array}$$

That is, the least diameter of stud should be 1⅞ inches in this. The thickness of the head *b′* should never be less than one half of the least diameter of the stud. The size of the eye will be shown in treating of the links of chains.

This completes the account of proportions for different parts of the ordinary pulleys ; but some remark is necessary as to the form of the groove in the periphery.

If a round rope of hemp or wire is used, the groove will be semicircular in section; if a flat rope, the groove will be rectangular; for an ordinary link chain it will be of the form shown in Fig. 6, the part *a b* of the groove receiving those links which lie flat, and the part *c* those on edge.

Fig. 6.

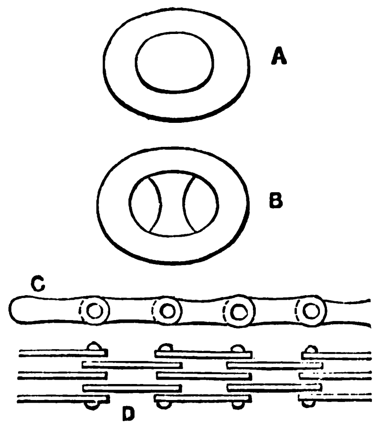


Of pulleys transmitting power we shall not treat in this place, as they belong properly to gearing, those pulleys with which we have here dealt merely doing the duty of altering the direction of a force.

We will now pass on to consider the practical points involved *in the manufacture* and use of chains of various

descriptions and in so doing we enter upon a subject which requires a very much greater amount of consideration than has yet been devoted to it in published statements; for every link of a chain has to transmit the whole strain to which the chain is subject, so that one faulty link renders the chain useless. Hence great care is necessary in testing a chain previous to applying it to the routine work for which it is designed, and, in point of fact, it is incumbent upon every engineer to examine carefully for himself any chain he may have made, notwithstanding its its having passed the scrutiny of the licensed testing house; for we have known a chain, certified as being tested up to sixteen tons, break a few days after with a load of only eight tons on it.

Fig. 7.



In the first place, we will consider the ordinary short link chain, of which one link is shown at A in Fig. 7.

If we could have a perfect chain, its strength should be equal to that of two rods of the iron from which the links are made; that is to say, a chain made out of iron one inch in diameter should bear in tension a load equal to that which two one inch rods would be capable of sustaining. In order to approach such a result, it is evident the the ends of the links should be "upset," or

thickened up, so as to present a sectional area approaching to the sum of the sectional areas of the sides of the link. Thus, it is observed, the ends of the links are when made of greater diameter than the sides.

Now, as to the actual strength of the manufactured chain, there is much to be said, and the causes of failure are several: first, the iron may be deficient in tenacity; second, it may be injured in the course of manufacture; third, the weld or shut may be bad; fourth, the gearing on which it is used may be unsuitable, and so bring unexpected and undue stress upon it.

Let us consider the theoretical compared with the actual strength of a chain. A piece of chain was made of iron one inch in diameter, of the best quality used in that manufacture, and of which the breaking strain was 25 tons per sectional square inch. The sectional area of one side of the link will be that corresponding to a rod having a diameter of one inch, or

0·7854 square inches.

Hence the sectional area of the *two* sides of the link will be double this, or

1·5708 square inches.

We may drop the last two decimals (08), and the breaking weight of the perfect chain would be

$$\begin{array}{rl} 1{\cdot}57 & \text{total area} \\ 25 & \text{tons per square inch.} \\ \hline 785 & \\ 314 & \\ \hline 39{\cdot}25 & \text{tons.} \end{array}$$

Thus we find the *theoretical strength* of the *perfect chain* should be nearly 40 tons; but the chain actually broke under a strain of 24½ tons gross load,—hence a chain of

good make may be assumed to have the same ultimate strength as *one* bar of the iron of which it is made.

It is a common practice to test chains up to two-thirds the breaking weight as a proof strain; but for ourselves, we see no use in going so high; and on the other hand are fully convinced by practical experience that many cases of failure are due to overstraining the material in the process of testing; for if the substance be strained up to such a point as to permanently stretch it, then it is permanently injured. The link of a chain will usually give way at the weld, hence every chain should be carefully examined before being used, in order to ascertain that the welds are perfect in every link. The method adopted consists in passing the chain through a smith's fire and heating it to redness, after which cold water is poured on each link: and if the shut be imperfect it will open, and thus exhibit the defect. At B, Fig. 7, is shown a stud link, which is the form generally adopted for heavy chain cables, the object of the stud piece or distance being to preserve the form and rigidity.

C and D show side elevation and plan of a portion of a flat link chain, such as is used for very heavy work. It consists of a series of short bars fastened together by pins or rivets, as shown. It will be observed that the links are arranged alternately in twos and threes; hence, to keep the strength of the chain the same throughout, those links which occur in twos should be thicker than those where there are three together. Thus we might alternate two links three-quarters inch thick with three links of half-inch metal.

The requisite size of the pins is determined by considering the strain on the chain and the number of places where *the bolt must be* sheared, together with the mode in which

such strain is distributed through the links. Where the three links occur, it may be assumed that the load is equally distributed through, or that there is one-third of the load on each of the three links; then it is evident that in the case of the outside link, the pin requires to be sheared in *one* place only for rupture to ensue; therefore we have the following simple rule:—

RULE.—*To find the greatest strain on one section of a pin in a flat link chain, divide the total load on the chain by the greatest number of links placed together.*

In the present case three will be the greatest number. Let the load be 15 tons, then the strain on one pin is thus found for one section in such pin.

$$\begin{array}{r} 3)15 \\ \hline 5 \text{ tons.} \end{array}$$

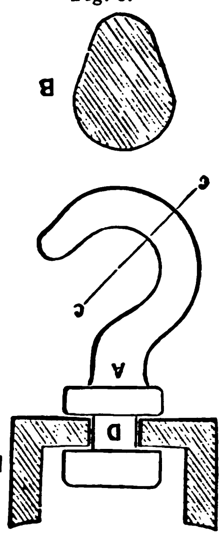
And as, to resist shearing strain, wrought iron is safe at 4 tons per sectional square inch, the area required will be 1¼ square inches, which area we find from a table of circles to correspond with a diameter of 1¼ inch nearly; therefore, for such a chain, the links should be fastened together with pins of that size.

In the next place we must determine the breadth of the links themselves. In the series of three we find on each link a strain of five tons; hence, as the *safe tensile* strength of wrought iron is five tons per square inch, one square inch of sectional area will be required effective. If the link be ½ inch thick, it must therefore be two inches broad; and to this must be added the width lost by the rivet holes, which is 1¼ inches; thus making the width over the eye altogether 3¼ inches. This width may of course be reduced in the central part of the link.

Where chains are merely running over pulleys, the links *should be kept as short* as may be, and the diameters of the

pulleys large, in order that a fairly continuous beaiing may be obtained for the links as they pass; for it is evident that a long link, in passing over a small pulley, will rest on the same at its centre only, hence the links will be subject to a bending or tranverse strain, for which chains are never designed. Where, however, such small pulleys cannot be avoided, flat places may be formed on the periphery to receive the links as they come round.

Fig. 8.



We will now pass on to consider the nature of hooks. At Fig. 8, A represents a hook carried in a shackle E, through which the neck D passes. B is a cross section through the lower part of the hook on the line *c c*. It is a species of egg oval, with the largest end on the inside of the hook, so as to furnish a good bearing for the ring, or other element placed upon the hook.

The portion of the hook D which affords means of attachment to the shackle E is proportioned in the same way as the neck *c* of the ring or eye bolt *b* in Fig. 5, and the part A of the hook is of course in tension only; hence it must have a sectional area of at least one square inch for every five tons of load to be carried. It may be useful to give some general idea for proportioning the sections of hooks in accordance with recent practice.

The radius of the large (or top) curve, which is a little more than a semicircle, should be twice that of the small or bottom curve, and the total depth of the section three

times the radius of the larger curve. The radius of the inside of the hook from the line *c c* backwards (that is, towards the left hand of the figure), should be one half of the depth of the section B. Then from practice we find the following rules.

RULE.—*To find the safe load on a hook* (*of the above proportion*) *in tons square, the depth of the section* (B) *in inches.*

Thus, the safe-working load on a hook 4 inches deep will be

$$\begin{array}{l} 4 \text{ inches deep} \\ \underline{4 \;\;\text{,,} \;\;\;\;\text{,,}} \\ 16 \text{ tons safe load.} \end{array}$$

This probably amounts to about $\frac{1}{5}$th of the breaking weight of a wrought iron hook. If the load be given we have—

RULE.—*To find the depth of section in a hook* (*of the above proportion*) *in inches to carry safely a given load in tons, take out the square root of such load.*

Thus the depth of hook to carry safely a load of nine tons should be three inches. These rules may be easily remembered if once the proportions of the sections be learned, as no fractions are involved in the calculation.

CHAPTER II.

WINCHES, CRABS, AND CRANES.

THE principle involved in the gearing of winches, crabs, and cranes, is the same in all, and in point of fact a crab is only a winch with additional gearing.

The object of gearing is to enable a small force moving at great velocity to overcome a greater force moving at a less velocity, or, on the other hand, to convert great pressure and low velocity, into a quick velocity with lower force or pressure. In any one machine the forces acting on any of the elements will be inversely proportionate to the velocities at which such elements are moving.

The simplest form of winch merely consists of a small drum, on an axis to which is fitted a handle at one end, or two handles one at each end. The gain in force over that applied by the handle depends on the ratio of the distance passed through in a given time by the handle to that passed through in the same time by the surface of the drum or barrel in which the rope is wound.

Let us assume for example that a winch has a barrel 6 inches in diameter, and a handle 15 inches radius. Now, because the circumferences of circles vary directly as their diameters, the relative distances passed through in a given *time (say one revolution)*, will vary as the diameters;

hence the distance through which the handle passes to that through which the surface of the drum passes, will be as 30 to 6 (for 15 inches being radius of handle, its end will describe a circle of 30 inches diameter). Hence, for a simple winch to find its force we have the following rule.

RULE.—*To find the pull which may be exerted by any given winch, multiply the sean force applied on the winch handle by one man, by the number of men working at the winch; multiply the product by twice the radius (or length) of the winch handle in inches, and divide the product by the diameter of the barrel in inches. The quotient will be the pull on the rope attached to the barrel.*

Let the average force which one man exerts in turning a winch handle be taken at 30 lbs., and assume the winch to have two handles, at each of which two men may work; then there will in all be four men working at the winch. Let the other dimensions be the same as taken above, the pull will be thus found:—

30 lbs., force of one man.
4 men working at winch.
120
30 ins. twice radius of winch handle.
Diam. of barrel 6)3600
600 lbs. pull on rope.

With regard to the speed at which such a weight would be lifted, we may assume that twenty-five turns of the handle may be made per minute; and of course the same number of turns of the barrel keyed on the axle or barrel shaft; then, as the circumference corresponding to 6 inches (the diameter of the barrel) is found to be 18·84 inches, that will be the length of rope wound up per revolution and the speed per minute will be

$$
\begin{array}{r l}
18{\cdot}84 & \text{inches per revolution.} \\
\underline{25} & \text{revols. per min.} \\
9420 & \\
3768 & \\
\hline
12)471{\cdot}00 & \text{inches per min.} \\
\hline
39{\cdot}25 & \text{feet per min.}
\end{array}
$$

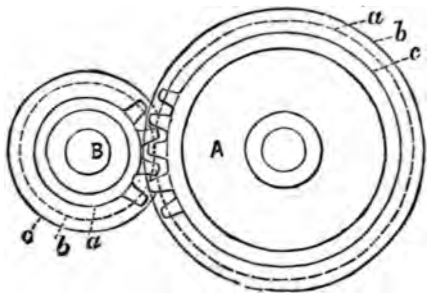
The winches however now in use increase the force in a much greater degree, there being spur gearing (toothed wheels) interposed between the handle and the barrel. In this case it will be necessary, in calculating the power, to determine the ratio of the *velocities per minute* of the handle and the barrel. Having found the relative numbers of revolutions per minute, we can then, from the ratio of twice the length of the handle to the diameter of the barrel, find the total increase of power; if in the above case, for instance, gearing were to be interposed between the handle shaft and the barrel, so proportioned that the handle revolves once for 100 revolutions of the barrel, the force would be increased by 100 times, or would be equal to a pull of 6000 lbs.; but as the barrel would only make one revolution while the men make 100, the speed would be lessened accordingly, and the speed at which the 6000 lbs. would be lifted, become 0·3925 feet per minute.

From this it will be seen that, by such machinery as that of which we are treating, there is *no increase of power*—for what is *gained in force*, is *lost in speed;* and power is a term used to express a certain amount of *work* done in a *given time;* and this is where, for want of a little consideration, a beginner may get confused (as perpetual motion schemers generally do) in trying to gain force without losing speed, or speed without losing force.

We will now pass on to consider the nature of the gear-

ing which is used to vary conditions of speed and force. The friction between surfaces of rollers in contact is such that, if there is no great opposition, the rotation of one may be communicated to another with which it is in contact; thus, we may imagine the dotted circles

Fig. 9.

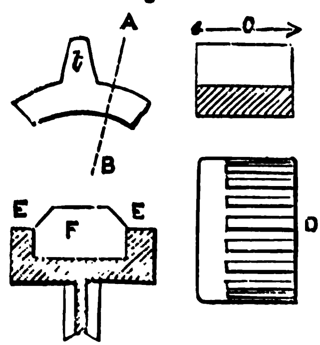


a, in the accompanying figure 9, to represent two rollers in contact and capable of revolving on their axes; then, if motion be imparted to one roller, it will drive the other the relative number of revolutions in a given time (or the angular velocity, as technical writers are in the habit of terming it) being inversely proportional to the circumferences (or diameters) of the rollers. Thus, if the roller A were twice as large as roller B, the latter would revolve twice while the former revolves once, for two revolutions of B would just enable it to travel once over the circumference of A. Usually the friction between the surfaces of the rollers would not be sufficient practically to overcome the resistance opposed to the motion of the wheels; hence the peripheries are serrated, or formed into teeth, a part being outside the surface of the supposed roller and part within, being placed between the circles *b* and *c*. The dotted circle *a*, at which the rollers would come in contact, if used, is known as the pitch circle. The teeth are somewhat longer inside the pitch line than outside, in order to

allow a little clearance, so that the teeth may not "lock" or jam. The distance between the containing circles *b* and *c* is called the length of the teeth, and their dimensions on the pitch line the thickness; the measurement parallel to the axis of the wheel, the breadth; and the distance from centre of one tooth to that of the next, measured on the pitch line, the pitch of the wheel.

In form, these teeth should be comprised between epicycloidal curves, in order to be mathematically correct; but it is a very common practice with millwrights to set them out with circular segments which nearly approach the true theoretical curve. The object of adopting a certain curve is, that the teeth may *roll on each other*, and not *rub their surfaces together* when working. As this requires, in addition to the exact curve, very true workmanship and exact "pitching," that is, placing the centres of the wheels at exactly their proper distance apart, it is not likely that perfect working is ever obtained, and, if it is at first, it will soon be lost by wear; hence it is not improbable that the circular arcs referred to are sufficiently near to give as good results as may be expected in general practice; the curves are generally struck above and below the pitch line in circular arcs, having a radius equal to, or nearly equal to, the pitch of the teeth.

Fig. 10.



Having said thus much about the form of teeth, we will pass on to consider a question of equal, if not superior importance,—that is,

the *strength* of the tooth of a wheel or pinion. Small wheels working in gear with large ones are commonly called pinions: *t* in fig. 10, is a side elevation of one tooth of a spur wheel, and *c* a front elevation of the same, viewed by taking a section at A B. The shaded part shows the ring portion of the wheel immediately below the teeth. The tooth is to be regarded as a short bracket fixed at one end and loaded at the other, hence the actual strain a tooth is capable of bearing is found by the following rule:—

RULE.—*The load that will break a sound cast-iron tooth is found by multiplying the square of the thickness of the tooth in inches by the breadth (c) of the tooth in inches, dividing the product by the length in inches, and multiplying the quotient by* 8000. *The result will be the breaking strain of the tooth in pounds.*

For an example, let us assume the tooth to be 3 inches long, 6 inches broad, and 2 inches thick at the base or root (where is the part at which it would break, being the section of maximum strain).

2 inches thick
2 inches thick
———
4 square of thickness
6 inches broad
———
Length 3)24
———
8
8000
———
64,000 lbs. breaking weight on tooth.

If we take one sixth the breaking weight as being the safe working load, then the wheel with these teeth may safely take a stress on the tooth of 10,666 lbs. In actual practice, however, we are required to calculate the thick-

ness of a tooth to sustain a given strain, rather than the strength of an existing tooth; and for this, of course, the thickness requisite *safely* to carry the load is required. Generally one sixth of the breaking weight is considered safe; and, on that supposition, we give the following rule:

RULE.—*To find the required thickness necessary to carry in safety any given stress in pounds, multiply such stress by the length of the tooth in inches, divide the product by the breadth in inches and by* 1333, *and extract the square root of the quotient; the result will be the required thickness in inches.*

Example: Let the stress be 10,000 lbs., and the breadth of tooth 6 inches, length 3 inches.

```
              10,000 pounds stress
                   3 inches length
Breadth   6)30000
       1333)5000(3·75
            3999
            10010
             9331
              6790
              6665
               125
               ...
```

We have now to find the square root of 3·75, which, from a table, we ascertain is 1·93 inches.

Some millwrights adopt as a standard proportion of breadth, to always make it twice the length, and by so doing the rule is simplified thus:—

RULE.—To find the breaking weight on a tooth, of which

the breadth is equal to twice the length, multiply the square of the thickness of tooth in inches by 16,000.

Let the thickness of the tooth be 2½ inches, required its breaking weight,

```
   2·5 thickness
   2·5 thickness
  ----
   125
   50
  -----
   6·25
     16000
  -------
  3750000
  625
 ---------
 100,000·00 lbs. breaking weight of tooth.
```

And to find the thickness to resist a given stress we have, taking one sixth of breaking weight as safe load:—

Rule.—*To find the thickness of tooth necessary to carry safely a given stress when the breadth of tooth equals twice the length, divide the stress in pounds by* 2666, *and extract the square root of the quotient. The result well be the thickness of tooth in inches.*

Let the stress be 16000 lbs.

```
2666)16000(6
     15996
     -----
         4
```

The square root of 6 is found to be 1·8 inches, which will be the required thickness of tooth.

The same laws herein given for spur wheels will equally apply to bevel spur wheels and worm wheels.

The usual proportions adopted for the teeth of wheels are here subjoined, forming what is termed the Manchester scale:—

hence, to find the strain on the wheel tooth, we divide the horse-power by the velocity, and multiply by the value given above; hence the rule:—

RULE.—*To find the stress on the tooth of a wheel driven by a prime mover of known power in lbs., multiply the horse-power transmitted by* 33,000, *and divide by the number of feet per minute the periphery of the wheel moves.*

The velocity of the periphery of the wheel in feet per minute is found by multiplying the number of revolutions of the wheel per minute by its diameter in feet, and by 3·1416; hence, embodying this in the foregoing, we obtain the following more complete rule:—

RULE.—*To find the stress on the tooth of a wheel driven by a prime mover of known power in lbs., divide the horse-power transmitted by the diameter of the wheel in feet, and by its number of revolutions per minute, and multiply the quotient by* 10,500.

Let the power transmitted be 40 horse-power, the diameter of the wheel 20 feet, and its number of revolutions per minute 5:—

Diameter of wheel 20)40 horse power

Revolutions per min. 5)2

0·4

10500 constant

2000

40

4200·0 lbs. on tooth of wheel.

From which, of course, the requisite dimensions of tooth may be determined by the foregoing rules.

As a rule, in a train of wheels commencing at one end *with a* pinion, that pinion drives a wheel on the shaft of *which is another* pinion driving another wheel carrying

another pinion, and so on, according to the number of motions used. In such a case, to find the ratio of volocities between the first and last shafts use the following rule:—

RULE.—*To find the ratio of number of revolutions (angular velocity) between the first and last shafts, multiply the diameters of all the wheels together, also multiply the diameters of all the pinions together, and divide the former product by the latter.*

Let us take an example of a train of wheels composed as under:—

First shaft carries a pinion 10 inches diameter, gearing with a wheel 36 inches diameter carrying a pinion 8 inches diameter, gearing with a wheel 40 inches diameter carrying a pinion 12 inches diameter, gearing with a wheel 60 inches diameter, fixed on last shaft.

Diameter of first pinion	10	36	Diameter of first wheel
„ second „	8	40	„ second „
	80	1440	
„ third „	12	60	„ third „
	960)	86400 (90	
		8640	
		0	

Hence the last shaft revolves once while the first shaft makes 90 revolutions; so we have a machine by which we can increase or diminish the speed imparted to it 90-fold.

Let us regard it as a crab, and on the first shaft there shall be a winch handle 15 inches long (describing a circle 30 inches diameter), and on the last there shall be a chain barrel 12 inches in diameter.

In one revolution of the first shaft the point of application (the end of the winch handle), passes through a circle corresponding to 30 inches diameter, that is in cir-

cumference 94·2 inches. The chain on the barrel passes in one revolution through the circumference of a circle 12 inches in diameter, or 37·68 inches; but the first shaft makes 90 revolutions to one of the barrel, hence the point of application of power passes through 8478 inches, while the rope passes through 37·68 inches; hence the pull is increased as follows:—

```
37·68)8478·00(225
      7536
      ----
       9420
       7536
       ----
       18840
       18840
```

Thus we see the pull is increased in the ratio of 225 to 1; so that on such a crab, one man pressing with an average force of 30 lbs. on the handle would be able to overcome a resistance 225 times as great; or,—

```
  225 ratio
   30 lbs. force applied
 ----
 6750 lbs.
```

Hence one man would be able to raise more than three tons, but of course very slowly.

We will now proceed to consider the proportioning of shafts, to resist torsion or twisting.

It is found that the strength of a shaft varies directly as the cube of its diameter, so the force being known that will twist a shaft one inch in diameter asunder, it is easy to calculate the breaking force of any other sized shaft of the same material. The force of torsion varies as the force applied multiplied by the length of the lever at the end of which it is applied; thus 120lbs. acting on a shaft through *a wheel 20* inches radius would produce torsion equal to *2400 inch lbs.*

Good wrought iron one inch in diameter will break under torsion of 12,000 inch lbs. Hence the diameter of a shaft corresponding to a given breaking weight will be found from the following rule:—

RULE.—*To find the diameter of shaft corresponding to a given breaking weight in pounds acting at a given distance in inches from the centre of shaft (for wrought iron), multiply the weight by the distance, divide by* 12,000, *and extract the cube root of the quotient.*

Let us assume that we require a shaft to carry safely a strain of 14,000 lbs. acting on the pitch circle of a wheel of 30 inches radius. For safety we will assume that we should not strain the shaft to more than one-fifth of its breaking load. Hence, if 14,000 lbs. be the safe load the breaking load will be—

14000 lbs. safe load
5 factor
70000

Hence we must calculate the diameter of a shaft, of which the breaking strain will be 70,000 lbs., acting at a distance of 30 inches from the centre of the shaft.

70000 lbs. breaking weight
30 inches distance
12000) 2100000 inch lbs.
175

The cube root of 175 is 5·59 inches, or say, $5\frac{5}{8}$ inches, and the diameter of the shaft should in no part be less.

For different materials the division 12,000 must be replaced according to the nature of material used. The following are the constants applicable to various shafts, round and square.

	1 in. round bar.	1 in. square bar.
Cast iron	11943	15200
English wrought iron .	12000	15360
Swedish " " .	11400	14592
Blister steel	20000	25497
Shear steel	20500	26112
Cast steel	21111	26880
Yellow brass . . .	5549	7065
Cast Copper . . .	4825	6144
Tin	1688	2150
Lead	1206	1536

As to the proportioning of shafts to resist dead weight or load upon them, if it acts near enough to the bearings to merely produce shearing strain, then the case may be treated in the same manner as explained for pulley spindles, and illustrated in Fig. 3. But on the other hand if a wheel or other load act upon a shaft at a distance from the bearings, it will give rise to transverse strain, and the following rule must be adopted:—

RULE.—*To find the diameter in inches of a shaft under cross strain with the load at the centre, multiply the load in pounds by length between bearings in inches, divide the product by* 14400, *and the cube root of the quotient will be the diameter of a shaft having the breaking weight equal to the given load.*

Let a wrought iron shaft be loaded with a wheel weighing 8000 lbs., its distance between bearings being 20 inches; it is required to find the diameter of a shaft to safely carry this load; let the strength of the shaft be five times the load, then the breaking weight should be

$$\begin{array}{rl} 8000 & \text{lbs. working load} \\ 5 & \text{factor of safety} \\ \hline 40000 & \text{lbs. breaking weight.} \end{array}$$

From which we thus find the diameter required.—

```
               40000 load in lbs.
                  20 in. between bearings
              ------
      14400 ) 800000 ( 55·5
              72000
              ------
               80000
               72000
               ------
                8000
                ····
```

The cube root of 55·5 is 3·81 inches, so in practice the shaft should not be less than 4 inches in diameter.

If, however, the load be not in the centre of the shaft, the length between bearings must be substituted by a number found from the following supplementary rule:—

SUPPLEMENTARY RULE.—*Multiply the distance of the load from one bearing in inches by four times its distance in inches from the other bearing, and divide the product by the length between bearings in inches.*

Let the length of shaft be 30 inches, and the load placed at 20 inches from one end, and therefore 10 inches from the other.

```
                20 inches distance
                10   „       „
               ----
               200
                 4  constant
               ----
   Length 30 ) 800
               ----
               23·3 inches.
```

Hence we should in this case use the previous rule, but substitute 23·3 inches for the length.

Before leaving the subject of wheels and shafts, a few

remarks upon keys for making the wheels fast on the shaft seem necessary. Wherever gearing is used to transmit power, there is a tendency of the wheels to turn upon the shafts instead of imparting rotatory motion to them; hence the wheels must be sufficiently firmly fixed on the shaft to prevent its slipping; and this is usually done by inserting one or more keys in a groove or grooves cut in the interior of the boss of the wheel and in the shaft, so that the thickness of the key is partly in the wheel, and partly in the shaft.

The effect of the stress on the wheel is to produce a tendency to turn the key over on its corner and split the boss of the wheel; hence it is necessary so to proportion the key that this tendency may be reduced to a minimum. The tendency to turn over will be at a maximum when the key is of perfectly square section and exactly half its depth in each; for then it is evident that if there be any weakness, the key will wear loose in *both shaft and wheel*, and in consequence may actually turn on its edge until it splits the wheel; but if the key be more deeply imbedded in the shaft it cannot turn round although the wheel should get somewhat loose, and in fact the key should be as firm on the shaft as if it formed a portion of its substance.

The actual nature of the normal strain on a key is shearing, as the wheel tends to shear off that part of the key which stands above the surface of the shaft itself; and in case of failure it would therefore be cut through a sectional area equal to its length, multiplied by its breadth. Thus a key four inches long, and three-quarters of an inch wide, would present a sectional area of three square inches requiring a force of about 50 tons to shear it through.

To find the stress on a key due to force transmitted through a given wheel, we have the following rule :—

RULE.—*To find the stress on any given key in pounds, multiply the stress in pounds on the periphery of the wheel by the diameter of the wheel in inches, and divide the product by the diameter of the shaft in inches.*

Let the stress on the periphery of a wheel 15 inches diameter be 25,000 lbs., and let the diameter of the shaft upon which it is to be made fast be 6 inches.

The stress on the key will be thus found,—

```
                  25000 lbs. on wheel
                     15 inches diameter
                 ------
                 125000
                 25000
                 ------
Diameter of shaft 6) 375000
                  ------
                  62500 lbs. stress on key.
```

The key should not be strained above one quarter of its breaking weight; hence the key should be calculated for a breaking strain.

```
 62500
     4
------
250000 lbs.
```

The ultimate shearing strain per square inch for wrought iron is 36,000 lbs. Hence the number of inches area required will be

```
36000) 250000(6·9, or say 7 inches.
       216000
       ------
        340000
        324000
        ------
         16000
         .....
```

So this strain would be met by one key 7 inches long by 1 inch wide. Of course the length of the key will be regulated by the thickness of the boss of the wheel.

Next we must consider the depth the key should be imbedded in the wheel and in the shaft, and we will commence by assuming the depth in the shaft to be one and a half times that of the key way in the wheel, so that three-fifths of the thickness will be in the shaft, and two-fifths in the boss of the wheel. The tendency will be to crush the metal in the boss of the wheel, and the ultimate resistance of cast iron to compressive force is 120,000 lbs. Therefore, taking the breaking strain as above at 250,000 lbs., the bearing of the key on the wheel should be

$$\begin{array}{r} 120000)\,250000\,(2{\cdot}08 \text{ square inches} \\ 240000 \\ \hline 1000000 \\ {\cdot}960000 \\ \hline {\cdot}40000 \\ \cdots\cdot \end{array}$$

Hence the depth of key way should be

$$\begin{array}{r} 7)\,2{\cdot}08 \\ \hline {\cdot}294 \text{ inches,} \end{array}$$

if only one key is used.

The strain on the shaft due to the key is partly tensile and partly compressive, but if the depth in the shaft be made from one and a half times to twice the depth in the wheel boss sufficient strength will be given.

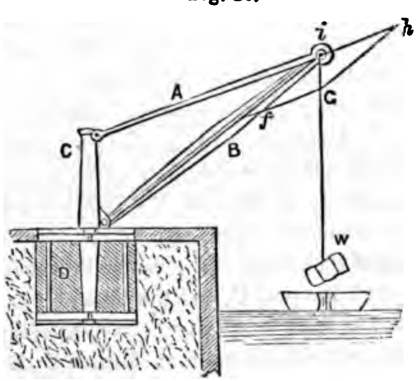
Where the strains to be transmitted are great, it is always advisable to have two or four keys in place of one, as by so doing the strain is more uniformly distributed round the boss of the wheel, and the liability to split is *reduced to a minimum.*

Having considered the proportioning of the various parts of the gear to withstand the strains to which it will be subjected, we will now pass on to consider its application to cranes, of which there are three principal sorts, post cranes, which are also called wharf cranes; wall cranes, also called warehouse cranes; and travelling cranes, which are mounted on wheels to render them portable.

Traversers are merely crabs mounted on movable platforms travelling on overhead girders.

Figure 11 represents an ordinary post crane, commonly used for landing goods. The gearing is not shown in the cut, as our present purpose is to deal with the main elements of the crane. It consists essentially of a post or pillar, C, which is firmly imbedded in masonry or other foundations, D, and to the pillar are attached the jib B and tie bar (or bars, there usually being two of them) A; the weight, W, is suspended from the end of the crane at the junction *i* of the jib and tie. We will now consider how these different parts should be proportioned.

Fig. 11.



The simplest method of finding the strains on the jib and tie consists in describing a parallelogram of forces upon the skeleton drawing of the crane, which is done as follows:—

From point i, where the jib and tie meet, a line is drawn in the direction of the weight W; being really a vertical line, the centre line of the tie is continued through i towards h. On the line i W mark off a convenient distance, representing to some known scale the load W, and let it be i G; then, from the point G, draw the lines G f, G h, parallel respectively to the lines i h and f i, then the lines f i, i h will represent the strains on the tie bar and jib respectively. That is, from the constructed parallelogram of forces f G h i, we find the strains by the following rules:—

RULE.—*To find the strain in lbs. on the tie bars, multiply the load in lbs. by the length* h i *in inches, and divide the product by the length* i G *in inches.*

RULE.—*To find the strain on the jib, multiply the load in lbs. by the length* i f (*or* h G) *and divide the product by the length* i G, *both in inches.*

G h and f i, being opposite sides of a parallelogram, are equal; hence, in the triangle G h i, we find the load and the two elastic resistances which sustain it represented respectively by the lines G i, G h, and i h, and the relative inclinations or dispositions of these are determined by the form of the crane. If a centre line were drawn vertically through the post C, and the tie and jib centre lines produced to meet it, we should have a triangle of exactly the same proportions as that formed by G i h. Hence, in calculation we may employ the triangle formed by the centre lines of the drawing instead of describing a separate parallelogram. The portion of the centre line of the pillar cut off between the tie rod and jib centre lines will be called the *height* of the crane, and the other lines, indicated as the lengths of the tie and jib, their lengths, being *measured from* their point of junction with each other to

their points of junction with the vertical centre line. The lengths may be taken in inches or feet, but must all be taken in the same name; if the height is in inches, so must be the jib, &c.

RULE.—*To find the strain in lbs. on the tie bars of a crane, multiply the load in lbs. by the length of the tie bar, and divide the product by the height of the crane; the quotient will be the required strain in lbs.*

Let the load on a proposed crane be 20,000 lbs., the length of tie bars (on centre line) 27 feet, and the height of crane post (on centre line) 8 feet, then we have

```
                   20000 lbs. load
                      27 feet length of tie bar
                  ------
                  140000
                  40000
                  ------
Height of crane 8)540000
                  ------
                   67500 lbs. strain on tie bars.
```

If we assume 10,000 lbs. per sectional square inch as safe load, then, dividing the above by this we get

```
10000)67500 lbs.
      -----
       6·75 square inches area of tie bars.
```

From this we find that two round bars, each 2⅛ inch in diameter, would meet the requirements of the case

Now for the jib we have

RULE.—*To find the strain in lbs. on the jib of a crane, multiply the load in lbs. by the length of the jib, and divide the product by the height of the crane, the quotient will be the strain on the jib in lbs.*

In the above case let the jib of the crane be 32 feet in length, the strain upon it will then be found from the foregoing rule as follows:—

$$\begin{array}{r l} & 20000 \text{ lbs. load} \\ & \underline{32} \text{ feet height of jib} \\ & 40000 \\ & \underline{60000} \\ \text{Height of crane } 8) & \underline{640000} \\ & 80000 \text{ lbs. thrust on jib.} \end{array}$$

If the jib be of cast iron, it may be considered safe to impose a strain of 12,000 lbs. per sectional square inch, so that its sectional area should be

$$\begin{array}{r l} 12000\ ) & \underline{80000} \text{ lbs. thrust} \\ & 6{\cdot}66 \text{ square inches.} \end{array}$$

If, however, it were made of wrought iron, the working thrust should not exceed 8000 lbs. per square inch, hence the area should be

$$\begin{array}{r l} 8000\ ) & \underline{80000} \text{ lbs. thrust} \\ & 10{\cdot} \quad \text{square inches.} \end{array}$$

The pulleys, bolts, and eyes for carrying same will be determined on the principles already laid down in the first chapter.

Having thus dealt with the ties and jib, we must consider the post or pillar. On this part of the machine it is evident there is a bending or transverse strain tending to break the pillar off close to the bottom where it is attached to the foundation, the strain being transmitted through the tie A to the top of the pillar C. We may, however, take a direct way to arrive at the stress. The load W is acting with a leverage equal to its distance from the centre of the pillar to bend the latter; so that, if, with the foregoing load, the distance from the centre of crane post horizontally to the line *i* W were 25 feet, the *moment* of bending strain on the base of the pillar would be

20000 lbs. load
25 feet leverage

100000
40000

500000 foot lbs. moment of strain.

If we assume that the working strain be taken at one fifth of the breaking strain, then the breaking strain of the post for the proposed crane should be

500000
5

2500000 foot lbs.

To determine the diameter corresponding to a given breaking load, we have the following rule :—

RULE.—*To find the diameter of a cylindrical beam (in inches) corresponding to a given breaking weight, multiply such weight in lbs. by its leverage in feet, and divide the product by* 480 *for cast iron (or by* 768 *for wrought iron) pillars (solid), and extract the cube root of the quotient.*

In the above case the load is 20,000 lbs., so the breaking load will be five times this, or 100,000 lbs. Suppose the pillar to be of wrought iron—

100000 lbs. breaking weight
25 feet leverage

768)2500000(3255
2304

1960
1536

4240
3840

4000
3840

160
...

The cube root of 3255 is 14·82 inches, hence we may say that practically the diameter of the pillar for such a crane should not be less than 15 inches.

Assuming the pillar is strong enough to withstand the bending strain, we have now to consider how the foundations stand affected. The pillar of the crane is firmly imbedded in a solid mass of masonry, D, which rests upon a bottom plate or radial casting, of which there is another on the surface. Then, supposing the foundation to be well made, and neglecting any assistance it receives from the surrounding soil in which it is imbedded, the weight will tend to turn it or upset it upon the edge nearest to the load, which load will act with a leverage equal to its distance from such edge; while to oppose it we have the weight of the foundation multiplied by the distance of its centre of gravity from the same edge. If, however, any actually bearing on the edge of the foundation were to occur, that edge would break off; hence, for safety, we will call the upsetting moment that which is equal to the load multiplied by its leverage measured up to the centre of the crane post.

Then the weight of foundation may be found corresponding to an overturning moment from the following rule:

RULE.—*To find the weight of a rectangular foundation in lbs. for a crane equal to a given breaking weight, multiply the load in lbs. by its horizontal distance in feet from the centre of the crane post, and divide the product by half the width in feet of the foundation.*

In the foregoing case let the foundations be made 12 feet square, then half the breadth will be six feet.

The load is 20,000 lbs., and using three as a factor of safety, the upsetting load would be 60,000 lbs., upon which we may proceed:—

```
  60000 lbs. upsetting load
     25 feet leverage
 -------
 300000
120000
 -------
6)1500000
 --------
  250000 lbs. weight in foundation.
```

Let the weight of the material used in the foundations be 120 lbs. per cubic foot, then, as the foundation is 12 feet square, each foot of thickness will weigh :—

```
  120 lbs. per cubic foot
  144 square feet
 ----
  480
 480
120
-----
17280 lbs. per ft. of thickness.
```

Whence we obtain the required depth of foundation by dividing the gross weight required by the weight per foot; thus :—

```
17 280)250000(14·4 feet
       17280
       -----
        77200
        69120
        -----
         80800
         69120
         -----
          1680
          ....
```

This foundation would, therefore, actually be made 12 feet square, and 14 feet 6 inches in depth ; but, of course, if made wider it would admit of being much reduced in depth, as not only would the weight per foot of thickness be increased, but also the leverage in favour of the foun-

dation. It may be interesting to illustrate the difference. Let the foundation be 18 feet square, the half of this will be 9 ft. So, proceeding as above, we find :—

```
      60000 lbs. upsetting load
         25 feet leverage
     ------
     300000
    120000
   --------
  9)1500000
   --------
     166666 lbs. weight in foundation.
```

From which it is observed that the total weight of materials required in the foundation varies inversely as the width of the foundation, so that, by widening the same, we economise masonry. Now, taking the weight as before, we find the weight per foot of thickness.

```
    120 lbs. per cubic foot.
    324 square ft. (square of 18 ft.)
    ---
    480
   240
  360
  -----
  38880 lbs. per foot of thickness.
```

Hence the thickness is,

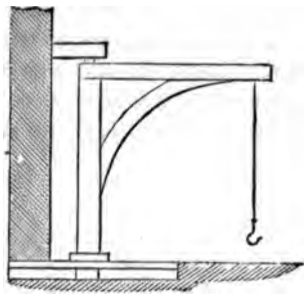
```
38880)250000(6·4 feet
      233280
      ------
       167200
       155520
       ------
        11680
        .....
```

Thus we find that the depth of the foundation varies inversely as the cube of its side (if made square, or of its diameter if made circular). Frequently, however, as in *cranes built on* wharfs, we are limited in space for the

foundations by a close proximity to the edge of the wharf. Hence, in such positions, foundations of considerable depth are sometimes unavoidable, but they should not be made deeper than the necessities of the case require.

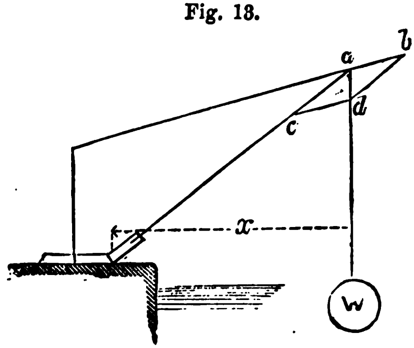
Figure 12 represents an ordinary wall crane, or warehouse crane. It is supported in a footstep at the bottom, and steadied by a strap at the top. The strain on the tie bar (which is generally horizontal) and on the jib may be calculated in the same way as in the post crane; but the crane pillar is not subject to similar strains, being, as it is, held at the top, and the jibs reaching nearly to the bottom, there is no bending strain on the pillar. The strain on the tie is taken as shearing strain on the top of the pillar, and transmitted to the strap which holds the top of the pillar.

Fig. 12.



In some post cranes, the lower end or base of the jib rests on a wheel supported on a roller path, on which it runs when the crane is turned round. In this case the strains on the

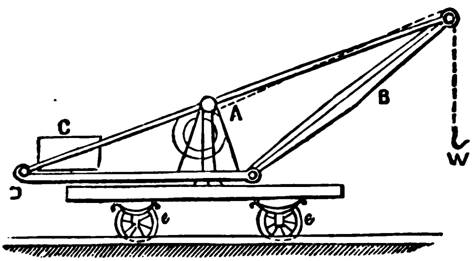
Fig. 13.



jibs and tie may be found from the parallelogram *a*, *b*, *c*. *d*, or by the dimensions of the crane, as previously described; but the strain on the pillar is proportionately less, the leverage of the weight being shown by the distance *x*, which is measured from the line of the weight W to the centre of the roller path.

For some purposes, as in erecting works, and in railway management, movable cranes are required, and these are arranged on trucks, so as to run on a permanent way. A crane of this sort is shown at Figure 14.

Fig. 14.



The machine is similar to a post crane, as shown at A B, W being the load, mounted on two pairs of wheels, *e e*, or, if the crane be very large, on three pairs. In this case it will be seen that the strain on the pillar cannot be taken by the foundation; therefore the load on the jibs of the machine is balanced by a weight, C, moving, which may be moved along the tail-piece D at the back of the crane, in order to adjust it to the weights to be raised. The weight C must of course be made sufficiently heavy to balance the heaviest load that may be picked up by the crane, the balance weight being at the extreme end of the tail-piece D, and this weight may readily be found from *the following* rule:

RULE.—*To find the weight in lbs. requisite in the balance box, multiply the load in lbs. on the jib of the crane by its horizontal distance in feet from the centre of the crane post, and divide it by the distance in feet of the centre of gravity of the balance box from the same centre.*

Let the load on the crane be 3000 lbs., its distance from the crane centre 10 feet, and the distance of balance box 4 feet; then the balance weight is

$$\begin{array}{r} 3000 \text{ lbs.} \\ 10 \text{ feet} \\ \hline 4\,)\,30000 \\ \hline 7500 \text{ lbs.} \end{array}$$

The tie, it will be observed, is continued past the pillar (being bolted to it, however), to the end of the tail-piece.

In order to prevent vibration on the springs, the truck may be either blocked up off the line, or shackles may be put on and the springs compressed on to blocks placed on the axle boxes; in this case, the crane is very steadily held, being clamped down firmly to the rails.

In arranging the details of framing, &c., for crabs and cranes, care must be taken to make any lugs, which there may be, sufficiently strong to carry the pull coming upon them. Thus there is a great strain on the lugs to which the tie rods are attached, and they will, in the generality of cases, be of cast iron. Let the tension of a rod be 20,000 lbs. The resistance of cast iron to tensile force amounts to only 18,000 lbs. Let the factor of safety be 6, then the breaking weight of the lugs would be

$$\begin{array}{l} 20000 \text{ lbs.} \\ \quad\;\; 6 \\ \hline 120000 \text{ lbs. breaking weight} \end{array}$$

Hence the number of inches sectional area required will be

```
18000)120000(6·66 square inches
      108000
      ------
       12000
       10800
       -----
        1200
```

Which, in practice, would be made 7 sectional square inches.

CHAPTER III.

HOISTS AND PRESSES.

Hoists may be generally described as a species of lift or crane with fixed jib, and always raising its load through some definite height or heights. The general form of hoists is too well understood to require an elaborate description; hence we shall herein touch only upon some of the details.

Hoists are of three kinds: those worked by air, called pneumatic lifts; by water, called hydrostatic lifts; and by steam. Air lifts are only suitable for low heights and light weights, and even then, except under peculiar circumstances, there is not much to recommend them. In principle an air lift is simply a float, and its rising is exactly in the same manner as the rising of a gas-holder as it fills. The air being prevented from escaping by the water (water-seal) around its lower edges, it is evident that the height of the surface of the water above the edge of the vessels rules the amount of pressure which may be maintained within the lift. Let us now see how a load is sustained by this air pressure. The lowest depth of water above the edge of the lift will be found when it is at the top of its stroke; hence this is all that is to be calculated upon. Let the lowest depth be two feet. Now every foot

depth of water corresponds to 0·434 lbs. pressure per square inch, hence two feet will give a pressure of 0·868 lbs. per square inch. Suppose the weight to be lifted be not more than 150 lbs.; then we thus find the area by dividing 150 lbs. by the pressure per square inch.

```
·868)150·000(172, say 173
      868
      6320
      6076
       2440
       1736
        704
        ...
```

173 square inches should, then, be the sectional area of the tube of the pneumatic lift in this case. We may, however, simplify the calculation, and find a rule to give at once the diameter of the tube or cylinder.

RULE.—*To find the diameter of the pneumatic cylinder in feet, divide the load in lbs. by the minimum depth of water above the edge of the cylinder in feet, extract the square root of the quotient, and divide by 7. The result will be the required diameter.*

Let the load required to be lifted be 500 lbs., and the depth of water 1·5 feet.

```
1·5)500·0(333·33
     45
      50
      45
       5·0
       45
        5
```

The square root of 333·33 is 18·3; hence we proceed:

7)18·3

2·61 feet, diameter of cylinder.

12

31·32 inches

Therefore its diameter should be 2 feet 7$\frac{3}{8}$ inches. The height of the lift is, of course, regulated by the length of the cylinder; thus, if the height required is 30 feet, the length of the tube must be somewhat more than 30 feet, —say 32 feet to allow for that part which is below the surface of the water when the lift is at the top of its range.

Hydrostatic hoists are worked by hydrostatic presses which may be actuated by pumps in the usual way, or by a head of water,—in which case, the apparatus may be distinguished by calling it a hydraulic lift. We will first speak of this class of lift, and assume the ram or plunger of the press to represent the whole length of the stroke, so that the table of the lift or hoist is placed directly upon the top of the piston.

Rule.—*To find the diameter of the piston in inches, divide the maximum load in lbs. by the available head of water in feet; extract the square root of the quotient, and multiply by* 1·73 *for the required diameter.*

Let the load to be lifted be 2000 lbs., and the available head of the water 50 feet.

50)200·0 lbs. load

40

Now the square root of 40 is found from a table of

square roots to be 6·32; hence we thus find the required diameter :—

$$
\begin{array}{r}
1{\cdot}73 \\
\underline{6{\cdot}32} \\
346 \\
519 \\
\underline{1038} \\
10{\cdot}9336
\end{array}
\text{ inches diameter of piston,}
$$

or the piston should be 11 inches in diameter.

In most cases this rule will be sufficient, as the given quantity will be the greatest available head of water.

In many hydrostatic hoists the stroke of the press is less than the lift, the difference being adjusted by pulleys and chains; but in this case the total pressure on the ram of the press must be much greater than the load to be lifted, in the same ratio as the height of lift is greater than the stroke of the press.

RULE.—*To find the load on press ram, multiply the load to be lifted in the hoist by height of lift in feet, and divide the product by stroke of ram in feet.*

Let the load to be lifted be 24,000 lbs., height of lift 40 ft., stroke of ram 6 ft. :—

$$
\begin{array}{rl}
 & 24000 \text{ lbs. load} \\
 & \underline{40 \text{ ft. lift}} \\
\text{Stroke of ram } 6) & 960000 \\
 & \overline{160000} \text{ lbs. load on ram.}
\end{array}
$$

And by applying the previous rule, the necessary diameter of ram to work this load can be found.

We will now proceed to consider the questions appertaining to the hydrostatic press, regarding, in the first [illegible], that class worked by hand.

The square root of 333·33 is 18·3; [illegible]

7)18·3

[illegible]

Therefore its diameter [illegible] ... [illegible] to the height of the lift is, of course, [illegible] attached to the cylinder; thus, if the [illegible] on pump length of the tube [illegible]

—say 32 feet to allow for [illegible] equal to 100 surface of the water [illegible] length to range. [illegible] fulcrum and

Hydrostatic [illegible] find the which may be [illegible] a head of [illegible] distinguished by [illegible] speak of this [illegible] of the press [illegible] so that [illegible] upon th[illegible]

[illegible] plunger.

[illegible] known, we [illegible] of the

[illegible] plunger [illegible] of the pump

[illegible] plunger [illegible]

```
                 10 inches
                 10
               ─────
                100 square of diameter of ram
               ═════
                 ·5
                 ·5
               ─────
                ·25 square of diameter of plunger.
               ═════
               2000 lbs. pressure on plunger
                100 ins. square of dia. of ram.
               ─────
Square of diameter
     of plunger   ·25)200000·00(800000 lbs.
                 200
               ─────
                  00000
                  .....
```

Hence the pressure one man would be capable of exerting, through the agency of such a press, is 800,000 lbs., or somewhat more than 300 tons.

Embodying the two rules now into one general rule, we have

Rule.—*To find the pressure exerted by a given hydrostatic press, multiply together the pressure in pounds on the end of the pump lever, the length in inches to the fulcrum of pump lever, and the square of the diameter in inches of the ram of press. Then multiply the distance from fulcrum of pump lever in inches to centre of pump plunger by square of diameter of pump plunger; and divide the former product by the latter, the quotient will be the total pressure the ram is exerting.*

We will assume the same proportions as above:—

```
 100 lbs. on end of lever
  60 inches length of do.
─────
6000
```

```
     6000
      100 inches square of diameter of ram.
   ------
   600000
     ·25 inches square of diameter of pump
      3  do.   distance of fulcrum from pump.
   ------
     ·75
     ·75)600000·00(800000 lbs.
         600
         -------
           00000
           .....
```

Being the same result as obtained by the other method.

Let us now assume that a press is required to exert a given pressure, that which can be obtained on the pump plunger being known, as, being derived from some certain source, as manual force or a steam engine, it is required to determine the relative diameters of the ram and pump plunger.

Rule.—*To find the relative diameters of press ram and pump plunger in order that a given pressure on the former may correspond to a given pressure on the latter, divide the required pressure on the ram by the given pressure on the pump plunger, and extract the square root of the quotient. The result will show the number of times by which the diameter of the ram must exceed that of the pump in order that the required pressure may be obtained.*

Let the available pressure on the pump plunger be 3,000 lbs., and the required pressure on the press ram 98,000 lbs.

```
     3000)98000
          ------
          32·666
```

```
   5)32·6660(5·71
     25
  107)766
      749
 1141)1760
      1141
       619
```

Hence, whatever size the pump plunger is, the press ram should be 5·71 times as large. Let the pump plunger be ·75 ($\frac{3}{4}$) inch in diameter, we thus find the required diameter of the ram

```
  5·71
   ·75 inch diam. of pump
  2855
 3997
 4·2825 inches diameter of ram.
```

In practice, the ram would be made $4\frac{1}{4}$ inches diameter.

So much for the power of presses. We will now proceed to consider the strain on the materials of which they are constructed.

As far as the ram is concerned, we do not require to make any calculation as hydrostatic presses are never worked to a sufficient pressure per square inch to endanger the ram, about 8,000 lbs. per square inch being the maximum, whereas in compression cast iron will safely bear 14,000 lbs. per square inch. Hence we have only to consider how the thickness of the metal in the cylinder is to be determined.

If the thickness of the metal in a cylinder is small in *proportion to its* diameter, the question assumes a simple

form, and the following rule will apply where the thickness will not exceed one-fifteenth of the diameter of the cylinder.

RULE.—*To find the thickness in inches of a cast iron cylinder of given diameter corresponding to a given breaking strain per square inch within, multiply the diameter in inches by the breaking strain, and divide the product by* 35000 *lbs.*

(NOTE.—The pressure per square inch in a cylinder is found by dividing the gross pressure on the ram by the area of the ram [that is, by the square of diameter in inches multiplied by 0·7854].)

Let the diameter of the cylinder be 8 inches, and the internal pressure to which it will be subjected 440 lbs. per square inch. Assume that the working pressure should be but one-fifth of the bursting pressure, we have

440 lbs. per square inch
5 factor
2200 lbs. per square inch

is the bursting pressure. Whence we find the thickness by means of the foregoing rule in the following manner:

2200 lbs. per square inch
8 inches diameter
35000)17600·0(0·5 inches thick
175000
10000
.....

Hence, in this instance, the cylinder only requires to be a full ½ inch in thickness.

Let us now apply this rule to thick cylinders, which are those subject to heavy internal pressure, and show wherein it fails. It is a most important matter to enter at some length into this question; because, as far as the author can ascertain, the existing formulæ published are fallacious, and

involve an absurdity; and, therefore, we will for a short space digress from our line of argument to show the type of rule that has been adopted. As a rule for thick cylinders of cast iron, we take the under one extracted from a very popular Engineer's Pocket Book.

(*a*) THE THICKNESS IS EQUAL TO HALF INTERNAL DIAMETER MULTIPLIED BY THE PRESSURE IN TONS, AND DIVIDED BY 7 LESS THE PRESSURE IN TONS.

Hence, when we want to get 7 tons pressure on the square inch, we find the divisor becomes 0; and, therefore, the quotient—that is, the thickness of metal—infinitely great; or, in other words, we could not make a press to work at 7 tons pressure to the inch.

Without digressing further, let us now apply the thin cylinder rule to the calculation of the thickness of a cylinder 10 inches in diameter to sustain safely a pressure of 4000 lbs. per square inch within it. Using the same factor of safety as before, the bursting pressure would be

$$\begin{array}{rl} 4000 & \text{lbs. per square inch} \\ 5 & \text{factor} \\ \hline 20000 & \text{lbs. bursting pressure.} \end{array}$$

Hence the thickness is thus found:—

$$\begin{array}{r} 20000 \text{ lbs. per square inch} \\ \underline{10} \text{ inches diameter} \\ 35000)200000(5{\cdot}7 \text{ inches} \\ \underline{175000} \\ 250000 \\ \underline{245000} \\ 5000 \end{array}$$

Or the thickness should, according to the rule, be 5¾ inches

nearly. This, however, would not be sufficient. We cannot compare this with the rule (*a*), as we have nothing stated about the factor of safety use; but we will investigate it independently, assuming for simplicity the thickness to be taken at 6 inches.

Under strain, metal, in common with other elastic bodies, is temporarily stretched or compressed, according to the intensity of the strain, and in a cylinder under internal pressure, the inner layers must be more strained than the outer, because the inner ones must be distended to press on the outer. Hence, the inner layer may be strained very much, whilst the outer one is scarcely strained at all; and in all cases the straining of the fibres in a given cylinder will vary inversely as their distances from the centre of the cylinder; and as action and reaction are equal and opposite, the resistance of each fibre will correspond to the point to which it is strained. When the breaking distension of the inner layer of fibres is reached, the vessel must give way; hence, if it be very thick, the mean breaking strain of the cylinder per square inch is less than the ultimate strength of a square inch bar under tension. When the cylinder is thin, as in the previous case, the rule holds good, as the difference of distension between the inner and outer layers of fibres is practically inappreciable; but in the present case it is different.

The breaking tension per square inch of sectional area on cast iron may be taken in even numbers at 17,500, and to find the tension on the outer layer of fibres, when the inner are at breaking point, we have the following:

RULE.—*To find the strain on the outer layer of fibres per square inch in lbs. corresponding to a given strain on the inner layer, multiply the given strain by the internal diame*

in inches, and divide the product by the external diameter in inches.

In the present case the inner diameter is 10 inches, and the outer, being the sum of the inner and two thicknesses of metal, is 22 inches.

```
                         17500 lbs. per square inch
                            10 inches inner diameter
                        ------
Outer diameter  22)175000(7954
                   154
                   ---
                    210
                    198
                    ---
                     120
                     110
                     ---
                      100
                       88
                      ---
                       12
```

Practically we may take the mean of these two amounts as the average resistance of the metal per square inch throughout the thickness.

```
  17500 lbs.
   7954 lbs.
  -----
2)25454
  -----
  12727 lbs.
```

Which may be assumed in round numbers as 12700 lbs.

We, therefore, find the internal pressure per square inch at which the inner layer of fibres is on the breaking point, by reducing the assumed bursting pressure in proportion to the ratio of 12700 to 17500, that is to say by multiplying *it* by 127, and dividing by 175. We assumed 20,000 lbs. *as the bursting* pressure.

```
           20000 lbs. per square inch.
             127
         -------
          140000
          40000
         20000
         -------
    175)2540000(14514 lbs.
         175
         ----
          790
          700
          ----
           900
           875
           ----
            250
            175
            ----
             750
             700
             ----
              50
```

Hence, the inner layers of the metal would be at their breaking stress when the pressure per square inch in the cylinder reaches 14,514 lbs., and the question remaining to be decided is what factor of safety should be taken. We may evidently take a much lower one than 5, as that was adopted as the assumption of all the fibres coming to the breaking stress at once, so probably 3 will be amply sufficient; then we have

```
    3)14514
    -------
       4838 lbs. per square inch,
```

as the safe working pressure in such a cylinder.

Much has been said about making hydrostatic press cylinders in such a way that, when cast, the outer layers

of metal contract upon and compress the inner; but the uncertainty here arises of knowing how much that compression may amount to, so that it can scarcely be taken into consideration.

It is a matter much to be regretted that there has been no series of experimental trials of thick cylinders upon which some satisfactory formulæ might be based.

We may now make a few remarks upon the modes of packing rams of hydrostatic presses.

The plan most generally adopted in England has been to use leather collars, into which the water obtained access under pressure, and so pressed the leather tightly against the cylinder on the one side, and against the ram of the press on the other, and thus preventing the escape of water. There is, however, a great objection to these leathers, inasmuch as they require to be renewed every few months; which, of course, causes some delay and expense.

This difficulty may be obviated by packing the rams with hemp in exactly the same way as the piston rod of a steam engine is packed, and the hemp will be found to wear out many leathers. We cannot say how long the hemp packing will last; but we know of one now that has been in use in an accumulator 18 inches in diameter, for three years, and it is not yet worn out.

Before leaving the subject of hydrostatic presses, it may be desirable to explain what an accumulator is, and the part it takes amongst hydraulic machinery.

Naturally the action of an hydrostatic press is exceedingly slow, as it stands to reason that a small force, being altered into an enormously large one (and the principle being admitted that the increment of force corresponds to the decrease of velocity; or, in other words, that in any *machine for altering* the intensities of forces, the intensities

of the forces at the opposite ends of the machines are inversely as the velocities with which those opposite ends move), that part of the machine which gives off the increased pressure must move at an exceedingly slow speed.

This want of velocity, besides causing waste of time, is also very strongly opposed to the use of hydrostatic pressure in cases in which it may otherwise be used with great advantage.

In order to overcome this difficulty, and get a quick motion together with great pressure, the accumulator has been contrived. Let us suppose that an hydrostatic press is required to be used once every ten minutes, and it would take say nine minutes to fill it so that it would make a complete stroke, and further suppose that it is required to act quickly. A slightly larger press is made, and its ram is loaded with a weight corresponding to the pressure required in the press which it is intended to work. Now if we assume that an engine is engaged constantly in pumping water into the accumulator except it stops when the accumulator is full, we shall get instantaneous action of the hydrostatic press; for, after the engine has worked nine minutes, there will be enough water pumped into the accumulator to make one stroke of the ram of the press. Self-acting gear stops the accumulator until the stroke of the press has been made when it is again started and filled so as to be ready for another stroke of the press, and thus the required end is attained.

Now it is evident that if the accumulator be made large enough it can be used for more than one hydrostatic press. And even if all the presses be not required to produce the same gross pressure, this can be attained by varying the diameters of the presses so that they may all work at the

same pressure per square inch, which pressure per square inch is ensured in the accumulator by a tank filled with weights or by means of weights suspended from a cross-head attached to the top of the ram.

We will now proceed to show how to determine the size of the tank or of the weights required for the accumulator.

Let us assume that a number of hydrostatic presses working at 1400 lbs. on the square inch, require to be furnished with an accumulator of which the ram is to be 20 inches in diameter: we have the following rule.

RULE.—*To find the weight in lbs. of the load on the accumulator ram, multiply the square of its diameter in inches by the pressure in lbs. per square inch and by* ·785.

```
          20 inches diameter
          20   "       "
         ----
         400 square of diameter
        1400 lbs. per square inch
        ----
      160000
      400
      ------
      560000
        ·785
     -------
     2800000
    4480000
   3920000
   ---------
   439600·000 lbs. load.
```

A cubic foot of cast iron of average quality weighs *about* 450 lbs.; hence the cubic feet of iron required to amount to the above weight will be thus found.

450)439600(976 say 977 cubic feet.
4050
3460
3150
3100
2700
400

A part of this weight is of course supplied in the ram of the accumulator. We will assume the length of the ram (including the head) to be equal to 15 feet. We will ascertain how many cubic feet it contains.

The diameter is 20 inches or 1·66 feet.

1·66 feet diameter of ram
1·66 „ „
996
996
166
2·7556 feet area of ram
15 feet length „
137780
27556
41·3340 cubic feet in ram.

This must be deducted from the above:

977	cubic feet,	gross required
41	„	in ram
936	„	required in tank.

Let the tank be made 10 feet by 10 feet by 9 feet clear inside measurement; that will furnish room for 900 cubic

feet. The rest being made up by the weight of the tank and its appurtenances.

We will now pass on to the last kind of hoist of which we purpose treating, namely the steam hoist, not however in this place entering into the calculations of the engines themselves, which will be duly considered in a subsequent chapter.

The chains are of various kinds as short-link chains, flat-link chains, flat wire ropes, &c., according to the loads to be raised, and the nature of the machinery used in the transmission of the power, this machinery amounting in fact to a steam winch in many cases.

If the lift is light, say under a ton weight, and not over 20 feet in height, it may be worked by a single-acting engine having a chain attached to a fixed pulley and rove through a grooved pulley, carried on the end of the piston rod, and thence taken over guide pulleys to the required position. The end of the chain will then travel twice as far as the piston, the gross pressure on which must of course be twice the weight to be lifted. Thus, if the lift be 14 feet, and the weight to be lifted 12 cwt., the stroke of piston will be 7 feet and gross pressure on it (excluding friction) 24 cwt. The mode of action is this: the piston is balanced at the top of its stroke by a counter weight attached to the platform of the hoist, which is at the bottom of the lift when the piston is at the top of its stroke. The platform being now loaded steam is admitted above the piston, and, forcing it down, raises the load. When the load is removed the steam is shut off, and that in the cylinder being allowed to escape, the counter-weight again draws the piston to the top of its stroke, ready for another lift.

The chief distinction between other hoists rests in whether they be driven by quick or slow running engines.

If slow running engines be used, the gearing will be of the same description as that used in ordinary winches, and may be calculated in the same manner, but if small engines running at high speeds be used, a very long train of spur gearing would be required ; hence it is advantageous to substitute the tangent screw and worm-wheel in its place, thus economising both space and cost.

Before proceeding to the mode of calculating the tangent screw arrangement, it seems desirable to discuss the objections which have been raised to it, and also the advantages claimed for it.

The first objection commonly raised is on account of the amount of friction which is involved in its use, the fact of the matter is, however, not that there is more friction in this arrangement than in the train of spur wheels, but the friction, instead of being distributed through a number of shafts is concentrated in one. Let us assume that gearing is required to reduce the velocity of the primary shaft to one-fiftieth in the final shaft. If we can place the two shafts at right angles, this can be done at once by the tangent-screw and worm-wheel arrangement, but if it be done by spur-wheels, a train of them will be requisite. Thus, say we have on the primary shaft a pinion 8 inches in diameter gearing into a wheel 40 inches in diameter, on a shaft carrying another 8-inch-pinion gearing into another 40-inch wheel on a shaft carrying a 12-inch pinion gearing into a 24-inch wheel on the final shaft; the end will be attained, as the final shaft will make one revolution for every fifty revolutions of the primary shaft. Here it will be observed we have four shafts amongst which the friction is distributed, whereas in the other arrangement there are but two, the worm-wheel shaft and the tangent screw shaft.

Theoretically there is no friction between the teeth of

spur wheels properly shaped and truly pitched, but *practically* there always is some friction; because, even if the wheels are in the first instance pitched dead true, the wear in the bearings will very soon impair the exactitude of their setting; and if the diameter of the wheel on the final shafts in both arrangements be the same, the strain on the teeth will be more in the spur wheel than in the worm-wheel; for of course the total strain will be the same at the pitch line of the wheel, but in the spur wheel the strain is all borne by one tooth, whereas in the worm-wheel the strain is spread over three or four teeth.

Whatever rubbing there is on the tooth of the spur wheel is radial, but the rubbing on the teeth of the worm-wheel is parallel to the axis of its shaft. We know by experience that the wear of a well-formed tangent screw and worm-wheel well constructed is by no means great if due attention be paid to its lubrication, but if this be neglected the worm will of course soon perish, the same as would a journal if dry.

There will be a thrust on the screw shaft in the direction of its length equal in intensity to the total thrust at the pitch line of the worm wheel, and assuming the worm wheel to be on the same shaft as the chain barrel, the thrust on the screw shaft may be determined as follows:—

Rule.—*To find the thrust on the screw shaft in lbs., multiply the strain on the chain in lbs. by the radius of the chain barrel in inches, and divide the product by the radius of the worm wheel in inches, the quotient will be the thrust on the screw shaft in lbs.*

Let the strain on the chain be 22,400 lbs., the radius of chain barrel 8 inches, the radius of the worm-wheel 18 inches.

```
                         22400 lbs. strain on chain
Radius of worm-              8 inches radius of barrel
      wheel . .  18)179200(9400 lbs. thrust on shaft.
                    172
                     72
                     72
                     00
```

To withstand which, the plummer blocks must be made sufficiently strong. If the strain be taken in tons the thrust will of course be in tons. Let the strain on the chain be 24 tons, the radius of the barrel 18 inches, and the radius of the worm wheel 11 inches.

```
                               24 lbs. strain on chain
                               18 inches radius of barrel
                              192
                              24
Radius of worm-wheel 11)432
                              39·3 tons thrust on screw shaft.
```

This thrust would have to be taken up by the two plummer blocks supporting the shaft on either side of the tangent screw, and sufficient bolts must be used to fasten the plummer blocks down to the framing to prevent any risk of their being sheared through. The total sectional area of the bolts will be found by dividing the thrust on the screw shaft by the resistance per square inch of wrought iron to shearing strain. Wrought iron *safely* resists 4 tons per sectional square inch; hence, in the present case, we find the total sectional area of the bolts to be

```
                        4)39·2 tons thrust
                           9·8 square inches.
```

If the bolts *used be 1* inch in diameter, the area of each

bolt in section will be 0·78 square inch; hence, dividing the total area by this, we ascertain the number of bolts required.

$$
\begin{array}{r}
\cdot 78)9\cdot 80(12 \text{ bolts} \\
\underline{78} \\
200 \\
\underline{156} \\
44
\end{array}
$$

The working shows a little over twelve bolts, but it is so little that we may practically with perfect safety neglect it, and consider twelve 1 inch bolts as the number required; that will then be six in each plummer block. The second example shows an enormous thrust due to the diameter of the chain barrel being so much greater than that of the worm wheel, whereas in the former case the worm wheel was much larger than the barrel in diameter, which is a far better arrangement; but we have inserted the second case because we have known of such apparatus being put up, although the use of larger chain barrels is altogether contrary to good practice, involving as they do very heavy gearing.

CHAPTER IV.

STEAM, AIR, AND GAS ENGINES.

Within the small space allotted to the chapters in the present treatise, it would be impossible to enter minutely into all the details of thermo-motive engines, and therefore we shall direct our attention more particularly to those points in which there is most chance of errors arising through either carelessness or ignorance. It behoves every one who buys an engine, and more especially one of the cheap class, to examine it thoroughly, or have it examined by some competent person; for, although any engineering firm of position would, for its own credit sake, try and see that no imperfect machinery left their shops, yet experience shows us that there are those who make engines simply to sell, and consequently knock them together as cheaply as possible, without much regard to efficiency.

We have known several cases of engines supplied with the valves so made that they would not, until altered, do one third of the work guaranteed,—and that too, within the last two or three years. We will cite one instance:

A pair of engines was ordered to work a crane, the weight to be lifted being twelve tons, the diameter of the cylinders and length of stroke being specified, and also the

velocity at which the engines were intended to be run. Now this information should be sufficient for any practical engineer to detail the engine, so that it should perform its work satisfactorily. For the pressure of steam being known, and the size of the cylinder, it is a simple matter to determine the size of the steam ports and passages. However, although the makers undertook to guarantee the performance of the machine, when set to work it would only lift four tons, and indeed barely that. In most respects the engines were tolerably right, only the steam passages were certainly too small; but the great error lay in the valves having a great excess of lap. The stroke of the engines was one foot, and the travel of the slide-valve $1\frac{3}{8}$ inches, and yet there was $\frac{3}{8}$ of an inch lap at each end of the valve; the steam ports, moreover, being only $\frac{7}{16}$ inch in the opening, measured parallel to the valve's line of motion. The engines were intended to run at a speed of from 100 to 150 revolutions per minute. Now, it will be seen that, with so great an amount of lap, the engine could not get its steam in any thing at all approaching its requirements, and consequently the valves were taken out and shortened, the steam ports being widened to nearly $\frac{9}{16}$ inch, and the valves so placed as to have something less than $\frac{1}{8}$ inch lap at each end. This having been done, the engines just did the stipulated work, but with nothing to spare; but, had the steam passages been properly proportioned, there is no doubt that fifteen tons might have readily been picked up. From such a case as this (and we know of several), it may be inferred how very necessary it is in doubtful cases to specify the size of steam ports, passages, &c., instead of trusting to the maker, and also to *examine the* valves *before* fixing the machinery in its place.

Very frequently—in fact we may say commonly—in

small engines the necessity for good, roomy steam passages is overlooked in the attempt to obtain a compact and elegant design, and it is certain that, to get good results, the steam pipes requisite for proper working do look disproportionately large. Having said thus much about the area of the steam ports, we will now give a rule, deduced from the laws of the flow of gases, by which the proper area may be determined under any given set of circumstances.

RULE.—*To find the area of steam passages in square inches, divide the absolute pressure in the cylinder (i.e., the pressure per square inch above the atmospheric plus* 15*) by the difference between the pressures in the boiler and cylinder, multiply the square root of the quotient by the speed of the piston in feet per minute, and by the square of the diameter of the cylinder in inches, and divide the product by* 15000.

From this we obtain the proper area of the steam passages for the particular cases that may demand our consideration. For safety take the minimum difference between pressures in cylinder and boiler with maximum pressure in the cylinder.

Let the diameter of cylinder be 20 inches, speed of piston 250 feet per minute, absolute pressure in boiler 40 lbs. per square inch, absolute pressure in cylinder 36 lbs. per square inch.

40 lbs. pressure in boiler
36 „ „ cylinder

Difference 4)36 lbs. pressure in cylinder.
9

We have now to take the square root of 9;

$$3)\dot{9}(3$$
$$9$$

and multiply the speed of piston,

250 feet per minute
3
———
750
400 square of 20 inches diameter of cylinder
———

Constant divisor 15000)300000(20 square inches.
30000
———
0

So we see the area of the steam port in this case should be 20 square inches—say 10 inches by 2 inches.

As a general practice, the exhaust port is made twice the area of the steam port.

From the 20 inches area we may determine the diameter of the steam pipe by the ordinary rule.

·785)20·000(25·4
1570
———
4300
3925
———
3750
3140
———
610

5)25·40(5·0 inches diameter of steam pipe
25
———
10) 40

The lap of the valve is determined according to the point at which it is desired to cut off the steam; but in small high-speed engines it should never be more than one-eighth of an inch at each end of the valve.

Some of the old engine builders used a set of cmpirieal rules for determining the areas of steam passages, &c., by making them proportionate to the areas of the steam cylinders; the constants were settled by observing what would answer well in practice, and so long as the steam pressure and speed of piston remain constant, of course such rules would answer well enough; but with the introduction of high pressure steam, the circumstances of each particular case are most materially altered, and it becomes necessary to take into the calculation all particulars of speed and pressure in order to obtain results of a satisfactory nature.

Having decided upon the amount of lap and lead to be given to the valve, the travel of the valve is at once determined by the following rule.

RULE.—*To find the travel of the slide valve in inches, multiply the width of the steam port in inches by* 2, *and add the lap in inches.*

Let the width of the port be 1·25 inches, and the lap 0·25 inches.

1·25	inches width of port
2	
2·50	
·25	lap in inches
2·75	travel of valve in inches.

It is a common practice in proportioning piston rods to make them one-tenth of the diameter of the cylinder, unless the steam pressure is very high when the following rule may be used.

RULE.—*To find the diameter of the piston rod in inches, divide the diameter of the cylinder in inches by* 55, *and multiply the quotient by the square root of the maximum pressure of steam in the cylinder in lbs. per* square *inch.*

Let the cylinder be 27·5 inches in diameter and the maximum steam pressure in the cylinder 36 lbs. per square inch.

55)27·5(·5
275   6 square root of 36
3·0 inches.

So the piston rod should be 3 inches in diameter.

The metal in the cylinder of a steam engine has two duties to perform; one is to resist the actual bursting pressure of the steam within it; the other, to withstand the vibration due to the motion of the machinery; and in addition to this, allowance must be made for wear. Including all, the following rule has been found satisfactory.

Rule.—*To find the thickness of metal in the cylinder in inches, multiply the pressure of steam in lbs. per square inch by the diameter of the cylinder in inches, divide the product by* 440, *and to the quotient add the square root of the diameter; divide the sum by* 8, *and the quotient will be the required thickness in inches.*

Let the pressure of steam be 25 lbs. per square inch and the diameter of cylinder 20 inches

25 lbs. pressure of steam
20 inches diameter of cylinder

Constant divisor 440)500(1·13   (*a*)
440
600
440
1600
1320
280
...

The square root of 20 is 4·47—

```
      1·13 first quotient (a)
      4·47 square root of diameter
    -------
    8) 5·60
    -------
       ·70 inches thickness.
```

This, which would be made ¾ inch, is the minimum thickness to be used.

The main shaft of the engine upon which is fixed the crank or crank plate, as the case may be, is proportioned to resist the torsion to which it is subjected by the following rule:—

RULE.—*To find the diameter of the main shaft in inches, multiply the horse-power of the engine by* 320, *divide the product by the number of revolutions of the main shaft per minute, and extract the cube root of the quotient.*

Let the horse-power be 40, and let the engine make 25 revolutions per minute.

```
                                320
                                 40 horse-power
                               -----
No. of revs. per minute  25) 12800(512
                              125
                              ---
                                30
                                25
                                --
                                 50
                                 50
                                 --
```

We have now to extract the cube root of 512, and by reference to a table of cube roots we find it to be 8 inches; hence, 8 inches is the required diameter of the shaft at its smallest part.

The next *point to* be considered is the manner of pro-

portioning the size of the cylinder to the work the engine is intended to perform, and for ordinary purposes engines are described as of so many horses-power. One horse-power is agreed to be equivalent to 33,000 lbs. raised 1 foot in 1 minute, and from this we can calculate the horse-power of any given engine thus:—

RULE.—*To find the horse-power of any given engine, multiply the mean pressure on the piston in lbs. per square inch by the square of the diameter of the piston in inches, by twice the length of the stroke in feet, and by the number of revolutions of the engine per minute, and divide the product by* 42000.

Let the mean pressure per square inch on the piston be 40lbs., the diameter of the piston 12 inches, the length of stroke 2 feet, number of revolutions per minute 30.

```
               40 lbs. per square inch mean pressure
              144 square of diameter of piston
             ----
              160
             160
             40
             ----
             5760
                4 twice length of stroke
            -----
            23040
               30 revolutions per minute
            ------
     42000)691200(16·4 horse-power
           42000
           ------
           271200
           252000
           ------
            192000
            168000
           -------
             24000
             .....
```

The speed of the piston in ordinary engines varies from about 200 to 300 feet per minute; so assuming 250 feet as the average speed, we can form a rule to find the diameter of cylinder corresponding to any given horse-power at a given pressure.

RULE.—*To find the diameter in inches of cylinder corresponding to a given horse-power, divide the horse-power by the pressure, extract the square root of the quotient and multiply it by* 13.

Let the engine be required to be 12 horse-power, the mean effective pressure in the cylinder be 20 lbs. per sq. in.

```
Effective pressure 20)12 horse-power
                      0·6

                  ·7)0·60(·77
                     49
               1·47)1100
                    1029
                      71
```

We have now to multiply this square root by 13:

```
                  ·77
                   13
                  231
                  77
                  10·01 inches diameter.
```

If the pressure taken be the gross effective pressure, then the corresponding horse-power is the gross horse-power, and is in excess of the available power, because a part of it is absorbed in overcoming the friction of the engine. The amount of power absorbed by friction varies according to the construction of the engine. Watt used

to consider one-third of the whole power as taken in working the condensing engine; but it will be quite safe to allow 25 per cent. for friction in condensing engines, and 15 per cent for high-pressure non-condensing engines.

It is usual to speak commercially of engines by their *nominal* horse-power, which in condensing engines is usually about one-third of the actual horse-power.

RULE.—*To find the nominal horse-power of a condensing engine, divide the square of the diameter of the piston in inches by* 30.

Let the diameter of the piston be 25 inches.

$$\begin{array}{rl} 25 & \text{inches diameter of piston} \\ 25 & \text{,, ,, ,,} \\ \hline 125 & \\ 50 & \\ \hline 30)625 & \text{square of diameter} \\ \hline 20{\cdot}8 & \text{horse-power, nominal.} \end{array}$$

RULE.—*To find the nominal horse-power of a non-condensing engine, divide the square of the diameter of the piston in inches by* 20.

Let the diameter of the piston be 8 inches.

$$\begin{array}{rl} 8 & \text{inches diameter of piston} \\ 8 & \text{,, ,, ,,} \\ \hline 20)64 & \\ \hline 3{\cdot}2 & \text{horse-power, nominal.} \end{array}$$

The nominal horse-power is, we may say, merely a term used in the buying and selling of engines.

Engines may be conveniently calculated by horse-power for such purposes as driving mills and machinery generally; for marine purposes and agricultural work; but in certain cases, as in winding engines and steam winches, it is more

convenient to calculate the diameter of the cylinder in another way.

Engines driving machinery, whether single or coupled, are usually fitted with a fly-wheel to equalise the motion; for when the engine is giving off more work than is being absorbed, the fly-wheel, with a slight increase of velocity, takes it up as accumulated work, and, at some other part of the revolution where the work done by the engine is below the average of the fly-wheel, parts with the excess of work which it had previously taken up.

RULE.—*To find the weight of the fly-wheel rim in cwts., multiply the total mean pressure on the piston in lbs. by the stroke in feet, and divide the product by the diameter of the wheel in feet, and by* 45.

(Note: the diameter of fly-wheel should be about four times the stroke.)

Suppose we have an engine with cylinder 10 inches diameter and 15 inches stroke, the pressure of steam being on the average 30 lbs. per square inch, the area of a circle of 10 inches is 78·5 square inches.

78·5 square inches area of piston
30 lbs. per square inch

2355·0 lbs. total mean pressure on piston
1·25 feet stroke

11775
4710
2355

Diam. of wheel 5)2943·75

45)588·75(13 cwts.
45

138
135

3

Thirteen cwts. will then be the proper weight for the rim of the wheel.

From this may be determined the sectional area of the rim of the wheel.

RULE.—*To find the sectional area of the rim of the fly-wheel, multiply the weight of the rim in cwts. by* 11·5, *and divide the product by the diameter of the wheel.*

$$\begin{array}{rl} & 11{\cdot}5 \text{ constant} \\ & 13 \text{ cwts. of rim} \\ \hline & 345 \\ & 115 \\ \hline \text{Diam. of wheel } 5) & 149{\cdot}5 \\ \hline & 29{\cdot}9 \text{ square inches sectional area.} \end{array}$$

In machinery for hoisting it is always necessary to have two engines acting on one shaft, the cranks being at right angles to each other, so that one or other engine always has a hold upon the load being lifted; for, with a single engine, there would be no getting over the dead point, except through the momentum of a heavy fly-wheel, which is not suitable where stoppages are sudden and frequent.

Now, it is evident that, with two engines so arranged, when one of them is on the dead point, the other is exerting its full force; but the engines must be made so large that one cylinder alone, when the piston is at mid-stroke, shall be capable of sustaining the load. To find the force which the engine must exert, we have the following rule:

RULE.—*To find the force in lbs. which must be exerted by one piston of a winding engine to hold a given load in pounds, multiply the given load in lbs. by the radius of the winding barrel in inches, and by the number of revolutions of the winding burrel per minute, and divide the product*

by half the stroke of the piston in inches, and by the number of revolutions per minute made by the engine shaft.

Let the load to be held be 22,400 lbs., the radius of the winding barrel 8 inches, the number of revolutions of the winding barrel per minute 3, the half-stroke 6 inches, and the number of revolutions of the engine shaft per minute 180.

22400 lbs. load
8 inches radius of barrel
179200
3
Half stroke 6)537600
Revolutions of engine 180)89600(497·7 lbs. force on piston
720
1760
1620
1400
1260
1400
....

We may call this 498 lbs. as the ·7 is continuous. The pressure on the piston must be equal to this. Assume the mean pressure per square inch on piston at 25 lbs.

25)498(29·92 square inches
25
248
225
230
225
50
50

Say 30 square inches for area of piston necessary to sustain load, and add one sixth for friction.

30	square	inches	for load
5	"	"	" friction
35	"	"	total area of piston.

From which we find the diameter—

$$\begin{array}{r} \cdot785)35\cdot000(44 \\ 3140 \\ \hline 3600 \\ 3140 \\ \hline 460 \\ \cdots \end{array}$$

The square root of 44 is 6·63 inches required diameter of cylinder; hence in practice the cylinders would be made 7 inches in diameter, and the stroke of piston 12 inches.

Let us now see what would be the mean force exerted by the two engines during the revolution, for it must be remembered that as the crank pin travels through the circumference of a circle having a diameter equal to the stroke of the piston, the piston itself travels only through twice the diameter of the same circle, making two strokes to each revolution. The ratio of the diameter to the circumference of a circle is as 1 is to 3·1416; hence the ratio of distances travelled between the piston and the crank pin is as 2 to 3·1416, because the piston travels two diameters while the crank pin passes once round the circumference. If we divide the latter by the former we shall have a multiplier for in any case finding from the mean pressure on the piston the mean force on the *crank pin.*

$$
\begin{array}{l}
3{\cdot}1416)2{\cdot}00000(0{\cdot}636 \\
\quad 188496 \\
\quad\;\; 115040 \\
\quad\;\;\; 94248 \\
\quad\;\;\; 207920 \\
\quad\;\;\; 188496 \\
\quad\;\;\;\; 19424 \\
\quad\;\;\;\; \cdots\cdots
\end{array}
$$

0·636 is then the constant factor which we may place in a general rule.

RULE.—*The mean total pressure on the piston of a steam engine being given to find the mean pressure on the crank pin, multiply that on the piston by* 0·636.

In the above case we find, excluding allowance for friction, that 498 lbs. is provided.

$$
\begin{array}{r l}
498 & \text{lbs. load} \\
{\cdot}636 & \\
\hline
2988 & \\
1494 & \\
2988 & \\
\hline
316{\cdot}728 & \text{lbs. on each crank pin} \\
2 & \\
\hline
633{\cdot}456 & \text{lbs. on two crank pins.}
\end{array}
$$

From which it will be seen that we have sufficient power for the work.

In respect to air engines we have not much to say, as they are not used to any considerable extent (in England at all events), and those that are in use are of small size. We must, however, point to a few matters connected with them of practical importance.

In the early air engines the difficulty most difficult to

contend with was the destruction of packing and working parts by the high temperature necessary to be used in order to obtain the requisite working pressure, and when we consider that by raising air 480 degrees, we only obtain an additional pressure of 15 lbs. on the square inch, it is evident that either high temperatures or very large machinery must be used. In order to keep the heat away from the working parts, the well-known plunger-shaped appendages are always attached to air engine pistons, except in the case of some few which, like Mr. Gill's, worked under water, the elastic fluid used being a mixture of heated air and steam.

During the past ten years another description of so-called air engine has come to some extent into use; the air is forced into the furnace which is closed by an air-tight door, and the air heated by passing through the furnace, and, *together with the gaseous products of combustion*, flows into the cylinder and propels the piston. There is in the principle of burning fuel under pressure an element of economy, as was practically established some years since, although it has not been applied to steam machinery; probably through the patentees of that method of working furnaces not having sufficient interest to press their invention; and no doubt what extra economy is exhibited by these air-engines in consumption of fuel may be traced to the close furnace, but these certainly have as much or more claim to be called gas engines as air engines; for all the gaseous products evolved from the solid fuel must aid materially in the propulsion of the machine. This, however, is not the place to enter more fully into a question which, further pursued, would become almost entirely *chemical.*

It is necessary in this class of engine to be careful what

description of fuel is used; for any kind of fuel which, during combustion, would evolve acid products, must necessarily cause rapid deterioration of the machinery by corrosion. The most durable packings for the working parts have been found to be those made of leather.

We will now pass on to gas engines properly so called. Lenoir's gas engine, which was one of the first, was actuated by exploding a mixture of air and gas at each end of the cylinder alternately, the ignition being effected by means of an electric spark. This contrivance cost more to work than a steam engine, besides being much larger in proportion to its power, and the only things that could be urged in its favour are, that it required no boiler, and there is no consumption when it is not working—indeed, this is the favourite claim of gas engine makers, that you do not require to keep a fire up, and an attendant to look after it, whether the engine works all day or only one hour a day.

In the Hugon gas engine the mixture of gas and air is exploded by gas jets, placed at the two ends of the cylinder, and the slides are so arranged that there is a communication between the mixture of air and gas in the cylinder and the lighted jet outside at the beginning of every stroke. As to the economy of this engine we cannot speak, having had no opportunity of testing it. The engines are bulky, as are all gas engines ; one of 3 horse-power being as large as a 10 horse-power steam engine.

CHAPTER V.

WATERWORKS, PUMPS, VALVES, STANDPIPES, ETC.

In designing pumps of large dimensions it is of great importance properly to detail the pipes and passages, and in all cases bends in the pipes should be avoided as much as possible. Wherever it is practicable, the suction and delivery pipes should be kept of diameter equal to that of the pump plunger; but if this cannot be arranged, they should be made as near as possible to that size, in order to reduce the friction of the water passing through the pipes as much as possible, and on the same principle in the designing of all details of water works, attention should always be paid that the friction of the water may be kept at a minimum. In speaking of the friction on water in pipes, &c., it is usual to treat the pressure on the water as the head of water producing such pressure; so, for instance, instead of saying that a body of water was under a pressure of 43·4 lbs. per square inch, it would be said that the body of water was under a head of 100 ft.; 100 ft. of water corresponding to a pressure of 43·4 lbs. per square inch, for a column of water 1 inch square on the base and 100 ft. in height weighs 43·4 lbs. For converting the head of water into pressure per square inch, and the converse, we have the two following rules:—

RULE.—*To find the pressure in lbs. per square inch due to a given head in feet of water, multiply the given head in feet by* 0·434.

Let the head of water be 174 feet.

174	feet head of water
·434	
696	
522	
696	
75·516	lbs. pressure per square inch.

RULE.—*To find the head in feet of water corresponding to a given pressure in lbs. per square inch, multiply the given pressure in lbs. per square inch by* 2·304.

Let the given pressure per square inch be 75·5 lbs.

75·5	lbs. pressure per square inch
2·304	
3020	
22650	
1510	
173·9520	feet head of water.

Having shown the relation between head of water and pressure, it will now be necessary to supply rules for determining the motion of water produced by such head or pressure.

The following rule serves to determine the size of orifice under a given head of water to discharge a given quantity per minute:

RULE.—*To find the area in square feet orifice to discharge a given number of gallons of water per minute under a given head of water in feet, divide such discharge by the square root of the height of the level of the water above the orifice and by* 1854.

Let the quantity to be discharged be 53,500 gallons per minute, and the head of water 16 feet. The square root of 16 is 4.

```
Square root of head  4)53500 gallons per minute
                     ----------
                  1854)13375(7·21 square feet.
                       12978
                       -----
                         3970
                         3708
                         ----
                          2620
                          1854
                          ----
                           766
                           ...
```

To determine the diameter of a pipe to convey a given quantity of water under a given head, we have the following rule:

Rule.—*To find the diameter of a pipe in feet to convey a given quantity of water in gallons per minute, divide the discharge by* 14,680, *square the quotient, and multiply it by the length of the pipe in feet; divide by the head in feet, and extract the fifth root of the quotient.*

Let the required discharge be 31,420 gallons per minute, the head of water 10 feet, and the length of pipe 70 feet.

```
14680)31420(2·14        (a)
      29360
      -----
       20600
       14680
       -----
        59200
        58720
        -----
          480
          ...
```

```
       2·14 quotient (a)
       2·14
      -----
       856
      214
     428
    ------
     4·5796 square of quotient (a)
         70 feet length of pipe
  ----------
10)320·5720
   --------
    32·0572
```

The fifth root of 32 must now be extracted, either by a process of logarithms or a table of fifth roots. Adopting the latter plan, we find the fifth root of 32 is 2; therefore the diameter of the pipe must be not less than 2 feet.

Note.—To reduce	cubic feet	to gallons,	multiply by	6·25
"	"	pounds,	"	62·5
"	gallons	" cubic feet	"	0·16
"	pounds	" "	"	0·016
"	gallons	" pounds	"	10·0

It may here be observed that the more exact weight of a cubic foot of water is 62·32 lbs.; but the data given above are sufficiently near for practice.

From consideration of the above formulæ, it will be seen that, in a vertical pipe, the friction is theoretically nothing, and the whole velocity due to the head should be obtained; but even admitting this, there is a reason why the suction pipe of the pump should be made as large as the pump plunger if possible, and that is, to allow the column of water to follow the rising plunger with a rapidity equal to its own motion, so that there may be no unnecessary delay in filling the pump barrel, and no whirling or com-

motion in the water entering. Now, if the suction pipe be half the diameter of the pump barrel, it is evident that, to keep the barrel full as the plunger rises, the water must leave the suction pipe with four times the velocity that the plunger is rising, and in consequence must cause eddies and vibration. It is, on the other hand, useless to have the suction pipe larger than the pump barrel, because the water would not come in any easier, as its ingress would then be limited by the area of the pump barrel itself.

It does not in the least signify of what material the pipes are made, so long as they are tolerably small for the water adhering to, or wetting, the sides of the pipe. The actual surfaces of friction are both water, as a layer of water lines the pipe, and the flowing water runs over this aqueous lining; so that, whatever material the pipe may consist of, the friction is between water and water.

Velocity is the great element in the friction of liquids, pressure in no way affecting it; but the friction varies as the cube of the velocity.

To find the delivery of single-acting pumps we have the following rule:—

RULE.—*To find the delivery per stroke in gallons of a single-acting pump, multiply the square of the diameter of the piston or plunger in inches by the stroke in feet, and divide the product by* 31.

This rule allows about five per cent. for leakage.

Let the diameter of the pump plunger be 20 inches, the stroke 8 feet.

20 inches diameter
20 " "
―――
400 square of diameter
8 feet stroke
―――
31)3200(103·2 gallons per stroke.
31
―――
100
93
―――
70
62
―――
8

It will, however, most frequently happen that the quantity to be delivered is given, and the diameter of the pump has to be determined. Suppose, for instance, a population of 40,000 required to be supplied with water by a pumping engine in a manufacturing town. Allowing 30 gallons per head per day, the total quantity to be pumped in 12 hours (not working the engine at night) will be

40000
30
―――
12)1,200,000 gallons per 12 hours
―――
60)100,000 gallons per hour
―――
1666 gallons per minute.

Let the engine be single-acting and make 8 strokes per minute:

No. of strokes 8)1666 gallons per minute
―――
208 gallons per stroke of pump.

The stroke of the pump being given, the diameter of the plunger may be found from this rule:

RULE.—*To find the diameter in inches of a pump to deliver a given number of gallons per stroke, multiply the given number of gallons by* 31, *and divide by the stroke of the pump; the square root of the quotient will be the diameter of the pump plunger in inches (allowing* 5 *per cent. waste).*

In the present case, let the stroke of the pump be taken at 10 feet.

208 gallons per stroke
31
208
624
Stroke of pump 10)6448
644·8

2)644·80(25·3 inches diameter,
4
45)244
225
503)1980
1509
471

The pump plunger would be made 25¼ inches in diameter.

With the single-acting Cornish pumping engines, commonly used in extensive water-works and for mining operations, the plunger of the pump is usually loaded with what is termed the preponderating weight, the action of the machine being thus,—the steam acting during the down stroke of the piston, the pump plunger is drawn up, and with it the preponderating weight, then the equi-*librium valve* being opened so as to equalise the pressure

on top and bottom of the piston, the preponderating weight presses the pump plunger down, forcing the water through the delivery valve to its destination. We will now show how the amount of this preponderating weight is to be calculated.

Let the water being pumped up be required to be raised 162 feet above the bottom of the pump plunger when at mid-stroke, the pressure per square inch corresponding to this weight is found by a previous rule.

```
   162 feet head
   ·434
  -----
   648
  486
 648
 -----
 70·308 lbs. per square inch.
```

Let the diameter of the plunger be as above 25·5 inches, then its area will be—

```
    25·5 diameter of pump
    25·5    ,,        ,,
   -----
    1275
   1275
   510
  ------
   650·25
     .785
  ------
   325125
  520200
 455175
 -------
 510·44625 square inches area of pump.
```

To get the total pressure to be overcome, and therefore

the amount of the preponderating weight, we multiply the area of the pump by the pressure per square inch.

510 square inches area
70 lbs. per square inch
35,700 lbs. preponderating weight.

In addition to this weight, more has to be added in order to overcome the friction of the engine, but this is usually ascertained by experiment; for in fact it is impossible to calculate to anything like accuracy the friction of an engine. When a pump is properly balanced the weight is in excess at the commencement of the stroke and deficient at its termination, gaining velocity during the first half of its stroke, and gradually coming to rest as the plunger reaches the bottom of its stroke.

The pressure of the water cannot again force the plunger upwards, as the closing of the delivery valve prevents its return.

It is a matter of vital importance that the pump valves should work accurately and quickly, as otherwise there will be a great loss of water. In some of the early pumps they used the clack or butterfly valves, but they expose so large a surface to the water *in proportion to their weight*, that they are very slow in closing, and two evils result therefrom; the first is great loss of water, amounting to as much as 16 or even 20 per cent. of the water; and the whole column of water descending with and upon the valve causes a very great shock, destructive to the machinery. We will take an example to illustrate the force of such a blow in a large pumping engine. There is an engine near London, the pumping plunger of which is 50 inches in diameter, the height of lift being 100 feet, this will represent

a pressure of 43·4 lbs. per square inch, acting on an area of nearly 2000 square inches.

$$\begin{array}{l} 43{\cdot}4 \text{ lbs. per square inch} \\ \quad 2000 \text{ square inches} \\ \hline 86800{\cdot}0 \text{ lbs.} \end{array}$$

This weight of 86,800 lbs. is equal to 38·7 tons. In the engine alluded to the valves are of excellent construction, so that any one standing close by can just feel them beat and no more; but if we assume that a flat valve had been used, at every stroke of the engine a blow equal to 38 tons falling through one foot would have been inflicted on the machinery.

The double-seated valves have been very much used for large pumps under high lifts, and give good results, and recently india-rubber valves have been employed with great success. We were first shown them in action by Mr. Morris, of the Kent Water Works, Lewisham, and we believe that gentleman was the first to introduce them. The valve consists in form of a series of cylinders diminishing upwards from the base pyramidically, and perforated with vertical slots, so that the valve altogether forms a series of cylindrical gratings, round each of which an india-rubber band is placed. The pressure of the water expands these belts, and passes them, after which they close noiselessly, the superincumbent pressure of the water keeping them perfectly tight.

Let us now consider the ordinary round valve used for small pumps, safety-valves, &c. It is necessary in fixing these to place some set screw or stop to prevent the valve being thrown out of its seat, and this stop should be fixed at such a distance, that it will suffer the valve to rise

just high enough to allow of the free passage of liquid through it.

In order that liquid may pass freely through a valve, it should open to such a height as will make the cylindrical opening under the edge of the valve equal to the area of the valve.

The area of the valve is equal to its diameter squared, multiplied by ·785. The area of the cylindrical opening is equal to the circumference of the valve, multiplied by the height to which it is opened, but the circumference of the valve is equal to the diameter of the valve multiplied by 3·1416. For the two areas to be equal, we find that the diameter multiplied by the diameter, multiplied by ·7854, should equal the height of lift multiplied by the diameter, multiplied by 3·1416.

Here in both quantities the diameter occurs as a multiplier, hence it may be cancelled in both; then the question stands thus: the diameter of the valve multiplied by 0·7854 is equal to the height of rise multiplied by 3·1416. Hence, if we multiply the diameter of the valve by 0·7854, and divide by 3·1416, we shall find the proper rise of valve; or as the height always varies in a certain proportion to the diameter of the valve, we can find that ratio by dividing ·7854 by 3·1416.

```
3·1416)·78540(·25
        62832
       ------
       157080
       157080
       ------
```

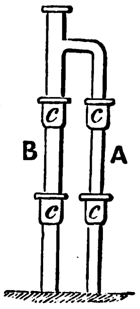
In some cases the water is pumped into a reservoir, in others the flow is regulated through a stand-pipe or air-vessel.

In pumping machinery furnished with a single acting

pump, the flow of water from the outlet valve is intermittent, the current being suspended during the stroke on which the pump barrel is filling with water; hence, if the water were suffered to pass direct into the pipes, it would flow through them in pulsations instead of continuously, wherefore it becomes necessary to interpose some contrivance to convert the flow into a continuous one.

An air vessel is usually made of spherical or cylindrical form, the water flowing in on one side and out on the other, both inlet and outlet being *near the bottom of the vessel.* When the water level gets *above* the inlet and outlet, it confines the air above, which then acts as a spring, the air being slightly more compressed as the downstroke of the pump plunger is made, and losing this extra compression during the upstroke, while no water flows in. These air vessels may be calculated according to the rule for their cast-iron cylinders from the pressure due to the head of water under which they are intended to work.

Fig. 15.



A standpipe is shown in Figure 15; A being the ascending pipe through which the water from the pump is forced, and from which it flows into the pipe B, at the bottom of which is an outlet to the pipes carrying the supply to its destination. The standpipe is built up in short lengths, being connected by the joints *c c*, which may be ordinary spigot and faucit joints, or united by bolts passing through flanges on the ends of the short lengths of piping.

As the water flows into the standpipe during the downstroke of the pump, the level of water rises slightly and falls again during the time when no water is flowing from the pump.

The thickness of metal may be calculated according to the rule given for thin cast-iron cylinders, the pressure being determined on any pipe from its distance below the surface of water in the standpipe.

From this it will be seen that the pressure on the pipe at the surface of the water is nothing; it there commences and reaches its maximum at the bottom of the standpipe; hence, in stand-pipes of great height, the upper pipes may be thinner than those at the bottom.

CHAPTER VI.

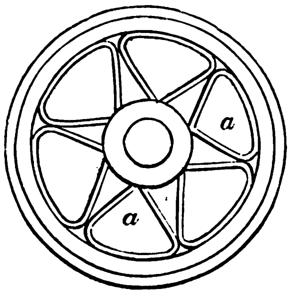
RAILWAY AXLES AND WHEELS.

WITH the constantly increasing weight of railway freight and higher speeds of travelling, the necessity of using wheels and axles of the most enduring quality augments. Even on the smoothest laid permanent way these parts of the rolling stock are unavoidably subjected to a series of shocks tending to set up a vibration most destructive to the materials of which they are constructed.

It may be as well here to allude briefly to the effects wrought by frequently repeated shocks on iron as regards the molecular condition of that metal. All, or nearly all, of the metals have a tendency, under favourable circumstances, to become crystalline in their structure, and slight agitations, insufficient to destroy cohesion, yet afford opportunities to the atoms to arrange or re-arrange themselves into crystalline molecules according to their affinities or polarity, and in the constantly repeated shocks which occur to the axles of a railway carriage or waggon are such agitations to be found. It is necessary under these circumstances that the axles be made in the best possible way, as the rupture of an axle is most likely to cause a train to leave the metals, thereby giving rise to the most serious results.

The best mode of manufacturing railway axles is by adopting the same mode of piling the metal in the forge as was used in Clay's large wrought-iron gun, which consists in laying together bars of a V section with all the apices melting, so as to build up a form approaching to a circular section, and from a pile so built up to roll the axles. By this arrangement a far more homogeneous section is obtained than when the axle is made from a pile of square bars.

Fig. 16.



In Figure 16 is shown a form of railway wheel in very common use, and which gives general satisfaction. The spokes, *a*, of this wheel are of wrought-iron, being formed of a series of bars bent into a form approaching a triangle, and placed together in the position shown, after which the boss of the wheel is attached. This boss may be either wrought or cast-iron. If a wrought-iron boss is used, the interstices at the inner ends of the spokes are filled with wedge-shaped pieces of iron, and the central portion of the wheel being raised to a welding heat, rings of wrought iron are placed centrally on each side of the wheel and welded up to the spokes and filling-pieces, thus producing a solid wrought iron boss in one piece with the spokes of the wheel.

If a cast iron boss be desired, the spokes fixed in position are placed in a mould and the boss is cast on; the cast iron, shrinking as it cools, takes a firm grip of the ends of the spokes, which are "nicked," or notched, to afford a better hold.

The boss of the wheel being complete the next process is that of "veeing up," which consists in welding in triangular pieces of iron in the spaces left between the shoulders or outer ends of the spokes, so as to form a complete circular perimeter to receive the tyre of the wheel. This is commonly done by hand, but by a modification of the steam hammer it might be effected with much greater precision and economy. The work, so far completed, is termed the body of the wheel, and when turned is ready to receive the tyre.

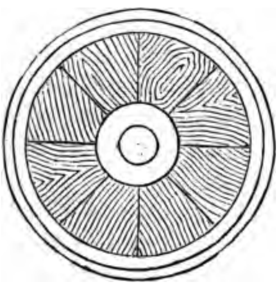
The tyre is bored out to a somewhat smaller diameter than that of the body of the wheel, and is *shrunk on*, being expanded by heat, until at a high temperature, its inside diameter is sufficiently increased to enable it to pass over the body of the wheel, on which it is then placed and allowed to cool; in cooling it contracts, and so takes a firm hold on the central part, after which it is further secured to the body by bolts or rivets passing through the tyre and that part of the body which lies between the spokes proper.

The amount of shrinkage should be only enough to cause the tyre to take a firm close hold round the body, for if it be too much, some part of the work must be more or less injured, and it is found that for the class of wheels above described, about one-sixteenth of an inch for every foot in diameter is sufficient; so for a three-foot-wheel the shrinkage allowed would be three-sixteenths of an inch.

The tyres are made in two different modes, according to whether they are to be welded tyres or solid, the latter being undoubtedly the soundest and best; the former are constructed of tyre bars bent into a circular form and shut or welded up the joint, being a "butt" joint, which is always awkward to make a good and secure shut with.

In making solid or weldless tyres, the following process is pursued. In the first place a cheese-shaped "use" is forged under the steam hammer, and subsequently a hole is punched in its centre, which is made sufficiently large for the rough forging so produced to be placed on a wheel rolling machine or "mangle." In this machine are two rollers, one of which is inside the tyre, and the other out, their distance being regulated by hydraulic or other pressure. On these rollers being set in motion, the circular use revolves, and from the pressure of the rollers gradually thins out, increases in so doing to the required diameter, and assumes the form of sectional area desired, and thus the tyre is formed without a weld.

Fig 17.

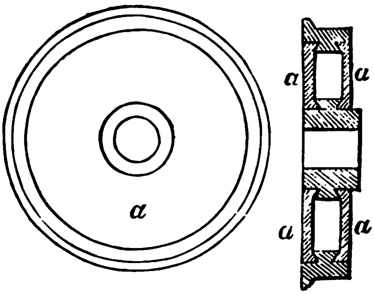


At Fig. 17 is shown a form of railway wheel known as Mansell's Wheel, the peculiarity of which is that the body of the wheel is made of wood instead of being of wrought iron as in the last case. The wood is retained between the boss and tyre by suitable plates and bolts, and in form is wedge-shaped, the various pieces being so arranged as to keep the wood in such a position that the grain shall radiate in all directions from the centre. This wheel has some great advantages, not the least of which is that they run with great smoothness, and by so doing decrease the vibrations of the carriage frame, reducing wear and tear, and adding to the comfort of passengers.

Figure 18 shows a section of wheel invented by Mr. Evans to obviate accidents arising from the tyre of the *wheel breaking*. The boss of the wheel is attached to

the tyre by means of two discs *a*, *a*, formed as shown, with a ridge or dove-tail round their peripheries, fitting into corresponding grooves in the tyre. Now by this mode of construction it is evident that even if the tyre should break up into several pieces, those pieces cannot separate from the body of the wheel because each portion is held in position by its own portion of the ridge and groove.

Fig. 18.



The number of patents that have been taken out for railway wheels is almost incredible, but we have not space here farther to dilate upon the matter.

CHAPTER VII.

TURNTABLES AND TRAVERSERS.

As in the cast of railway wheels, so with turntables, the great importance of adopting accurate principles in design and construction cannot better be attested than by the number and variety of plans which have from time to time been developed and patented, each one asserting some special superiority over its predecessors.

Turntables are used for two purposes: to enable a carriage or engine to be transferred from one track to another, laying at an angle to the track from which such engine or carriage is coming; and, secondly, for turning end for end such engines and carriages as are desired always to run with the same end forwards, this being desirable with all locomotive engines having tenders, and with some carriages peculiarly fitted in the axle boxes.

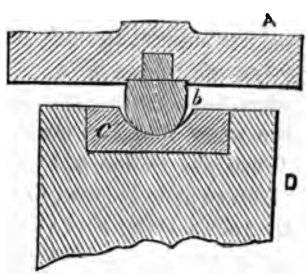
The principal or essential parts of which all turntables consist are as follows:—Commencing at the foundation, we have first the centre pillar or the roller path, or as it is technically termed, the roller race. These two parts sustain the whole weight of the turntable, and its superincumbent load between them, in proportions dependent upon the design of the superstructure, and they are firmly secured to the masonry foundations provided to receive them.

The superstructure consists mainly of the turntable proper, or platform, provided with a centre upon which to revolve, and having underneath its periphery the rollers upon which it rests, and which, when in motion, travel upon the bottom roller race. The turntable top is provided on its under side with rails resting on the rollers, and the rollers themselves are kept in position by radial rods and rings, the former preserving their proper distances from the turntable centre, and the latter their correct distribution round the circumference of the table. From this arrangement it will be observed that the table top must make two revolutions for the rollers to travel once round the roller path.

Upon the top of the table are fixed rails upon which may be run the carriages or engines required to be turned.

It has been proposed to replace the peripheral rollers under the turntable by balls or spheres running in a grooved race, but, as might have been anticipated, the result was a failure; for as it would be practically impossible to obtain all the spheres for any particular table of exactly the same size, so it follows that within a short time some of the spheres overtake the others, and becoming thus jammed together, require great force to move them, and the table becomes then useless until it is readjusted, when the same course is again followed hence these tables have not come generally into use.

Fig 19.



In figure 19 is shown an ordinary pillar or centre for a turntable. A is called the cap, and to it is attached the turntable top or plat-

form on the underside of the cap, and in its centre is placed the "centre," *b*, which should be made of steel; it is hemispherical on its under end, which rests in a cup, *c*, having the recess in it slightly larger in radius than is the hemispherical end of the centre. D is the body of the pillar.

In tables of the above description the greater portion of the load is assumed to be supported by the rollers, and this evidently involves a great amount of friction which, by adopting a different principle, may be avoided. The principle referred to is that used in Sellers' table, which is very largely applied for locomotive turntables.

These turntables being required to take a locomotive and tender and turn them end for end, need only one track of rails, but require to be about forty feet in diameter. They are usually formed of two wrought iron plate girders, one under each rail; they are braced together transversely, and bolted to a jacket attached to the cap of a centre similar to that described above and shown in Fig. 19. Under the table, at its outer ends, are rollers, two at each end, and under them a roller race running all round the turn-table pit. These rollers, however, are not intended to sustain any load while the table is being turned, but only to support the ends of the table while the engine is being run on and off it. They are, therefore, so fixed that when the table is perfectly level they are not resting on the roller race but stand about half an inch above it. When the engine and tender are run on to the table properly, so as to balance, the whole weight is upon the centre, the rollers being above the rails; hence the load is very easily moved. A forty-foot turntable loaded with fifty tons equally distributed may be readily turned round by one man.

In making the flooring of turntables formerly wood was principally used, but more recently its place is being taken by iron chequered plates, which form a strong and durable platform.

The pillar carrying the centre of the turn-table is, of course, in compression, and the bolts attaching the table-top to the cap in tension.

Let us assume a turntable to weigh 14 tons, and its maximum load to be 50 tons, this will make a total gross load of 64 tons on the centre.

Assuming cast iron to carry 7 tons per square inch safely, the sectional area of the smallest part is thus found:—

$$7\overline{)64}\text{ tons load}$$
$$9{\cdot}1 \text{ square inches.}$$

Which would correspond with a diameter of $3\frac{3}{8}$ inches. In practice, however, they are never made so small, the size being judged of as a matter of proportion rather than direct strength.

Taking the safe load on the bolts at 5 tons per sectional inch we find—

$$5\overline{)64}\text{ tons load}$$
$$12{\cdot}5 \text{ square inches.}$$

So the gross area of the bolts must not be less than 12·5 square inches. If bolts 2 inches in diameter be used, the effective sectional area will be 3 square inches in each bolt. Hence the number of bolts will be—

$$3\overline{)12{\cdot}5}\text{ square inches}$$
$$4{\cdot}16 \text{ bolts,}$$

or somewhat more than 4 bolts. Now in this case we need ample strength, for it must be remembered that the whole

load is hanging on a very few bolts, and the failure of one of them would be a very serious matter, losing at once one quarter of the whole strength of the apparatus. Hence, in the present case we should put in 6 bolts, each 2 inches in diameter, and in addition to this, to guard against defect in the thread of the screw, an additional nut, or lock-nut as it is called, is put on over the ordinary or working nut.

The girders which carry the table-top are placed directly beneath the rails, and of course in tables with more than one track they intersect beneath those points where the rails on the top cross each other.

We may now make a few remarks upon the subject of carriage traversers. The object of a carriage traverser is to take a carriage or waggon up from one line of rails and carry it laterally to another, and there lower it on to the metals. Rails are laid at right angles to the ordinary metals for the traverser to run upon, being of course cut away at parts where they would interfere with the passage of the flanges of railway wheels. There is nothing in the details of these frames requiring special notice.

CHAPTER VIII.

SWITCHES AND CROSSINGS.

In the manufacture of switches and crossings required for permanent way, the greatest care is requisite; as inaccuracy of design or construction may lead to most serious accidents. For instance, if the "tongue" of the switch should not close properly upon the back or main rail so that the point of the tongue fails to catch the flange of the leading wheel of the train, then it may happen that the train will run in between the metals on to the ballast, and thus be thrown off the track altogether; and if the tongue rails be made with thick points projecting beyond the top of the main rail in a lateral direction, the wheel flange striking may mount on the metals and so run off; this last form of switch tongue is, however, scarcely ever made now.

Fig. 20 shows the general form of the tongue rail and back rail of an ordinary switch. A, B, shows the back or main rail with the tongue C, D, closed upon it, so that any wheel travelling on the main rail in the direction from B to A, would be led off by the point D of the tongue rail, and would run on the tongue from D to C, and thence to the metals following. *a*, *b*, and *c*, *d*, show the same switch when open, in which position a wheel proceeding from *b*,

towards a, would pass on along the main rail unaffected by the tongue c, d. We have only shown the one tongue and back rail to illustrate our remarks as to the method in which the tongue should be fitted; the further *general* arrangement not being contemplated in a work on details.

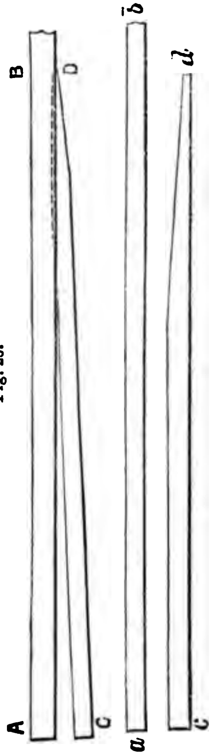
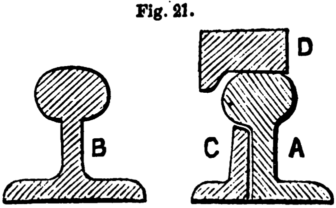


Fig. 20.

In fig. 21 are shown sections of a switch from which the form required to show the "housing" of the tongue under the main rail may be observed. A is the main rail of a general section, B, but partly cut away on the bottom flange to allow of the closing of the tongue C. D shows the tyre of the wheel. Now the section A, is taken close to the end of the tongue, and where its extremity is completely under the head of the rail A, from which it will be observed that the flange of the wheel will pass on and cover the point of the tongue rail while the wheel is yet on the back or main rail A, and as the tongue rail curves away from the back rail it will gradually lead the wheel by its flange into the track with which the tongue rail communicates. The tongue rail *is gradually* tapered from

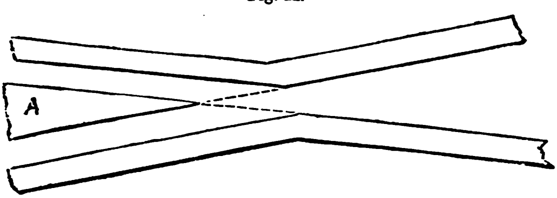
Fig. 21.



its full section down to that shown at C, being also curved so as to properly fit at its extremity; for if the extreme end of the tongue should stand away from the back rail, the flange of the wheel might pass between the two and thus run off the track; and if the point of the tongue were so thick as to project beyond the side of the head of the rail, the flange of the wheel might mount it. It does not follow as a matter of course that such results must accrue upon the defects alluded to, especially on a straight track, as there is always a certain amount of play allowed between the wheel and rail, which may be quite enough to clear the over-hang of the point of the tongue; but still, there is always the danger, and more especially if the point be located on a curve.

Crossings are required where one line of rails intersects another, and are of the general form shown in fig. 22, though

Fig. 22.



of course the angle varies, according to circumstances. The part requiring most attention as being most liable to wear is the point A; and for the construction of this portion, a great many different designs are in use. Formerly it would be made of two pieces of rail planed down and welded together so as to form a solid point; and subsequently the crossings have been made in solid steel. The points have also been formed by housing the end of one rail under another, and there firmly securing it; the rail to which

the housed rail is attached having its extremity planed away to form the point. This mode of construction affords the opportunity of taking the crossings to pieces if required; but we are rather in favour of solid crossings, if well formed and carefully executed: and, certainly, in all cases we should use solid points, whether of steel or iron; adopting in preference the former material.

CHAPTER IX.

BOILERS, TANKS, AND GIRDERS.

The work performed in the boiler yard comprises the construction of all kinds of works, consisting principally of plate and bar iron riveted up, and thus includes boilers, tanks, gasholders, evaporating vessels, girders, &c. In boilers, &c., it is necessary that the joints should be sufficiently close to effectually prevent the passage of liquids or fluids under pressure; hence, in these cases, greater care is required than in the riveting up of bridge or girder work. In the first place, it is desirable to know the size of rivet best suited for any particular thickness of plate, always remembering that in boiler work we have but two thicknesses of plate to rivet together, except at the corners of plates in some kinds of boilers, but in such places the plate is hammered down thin, or rather, tapered, in order to avoid spaces which would require caulking.

The following table shows the diameters of rivets suitable for different thicknesses of plate, as found in practice. The table also shows the proper lengths for countersunk and snap-headed rivets.

THICKNESS OF PLATE.	DIAMETER OF RIVET.	LENGTH COUNTERSUNK.	LENGTH SNAP-HEADED.
$\frac{1}{4}$ inch.	$\frac{1}{2}$ inch.	1 inch.	$1\frac{1}{4}$ inch.
$\frac{5}{16}$	$\frac{5}{8}$	$1\frac{1}{8}$	$1\frac{1}{2}$
$\frac{3}{8}$	$\frac{5}{8}$	$1\frac{1}{4}$	$1\frac{5}{8}$
$\frac{7}{16}$	$\frac{5}{8}$	$1\frac{3}{8}$	$1\frac{3}{4}$
$\frac{1}{2}$	$\frac{3}{4}$	$1\frac{1}{2}$	2
$\frac{9}{16}$	$\frac{3}{4}$	$1\frac{11}{16}$	$2\frac{1}{4}$
$\frac{5}{8}$	$\frac{3}{4}$	$1\frac{15}{16}$	$2\frac{3}{8}$
$\frac{11}{16}$	$\frac{7}{8}$	$2\frac{3}{16}$	$2\frac{5}{8}$
$\frac{3}{4}$	$\frac{7}{8}$	$2\frac{3}{8}$	$2\frac{3}{4}$
$\frac{13}{16}$	$\frac{7}{8}$	$2\frac{1}{2}$	$2\frac{7}{8}$
$\frac{7}{8}$	1	$2\frac{5}{8}$	3
$\frac{15}{16}$	1	$2\frac{3}{4}$	$3\frac{1}{8}$
1	1	$2\frac{7}{8}$	$3\frac{1}{4}$

The pitch of rivets in boiler work should be as close as practicable without injuring the metal, for if the rivets be too close, cracks will occur between them in the line of centres, thus destroying the strength of the joints. For work having $\frac{3}{4}$ inch rivets snap-headed, $1\frac{3}{4}$ inch pitch may be used with sound boiler plates of the best quality, but with ordinary ones it is not advisable to adopt less than 2 inch pitch, and the same ratios of pitch to diameter may be used for other sizes of rivets.

The strength of a boiler is calculated in a manner similar to that adopted for pipes; but we have to consider the strength afforded by different modes of construction, and also to observe which parts of the boiler are under tensile, and which under compressive strain.

In a tubular boiler, such as a locomotive boiler, we find tension on the outer shell, and compression on the tubes through which the flames and heated products of combustion pass from the fire-box to the smoke-box.

In a tubulous boiler, such as Craddock's or Jordan's, where the water is in the tubes and the fire outside them, we find tensile strain only, and in such boilers as are formed of flat plates stayed together, similarly to the fire-box and casing of a locomotive, there is a transverse or cross breaking strain brought upon the plates, and a direct tensile strain upon the tie bolts which are necessarily used to prevent the bulging asunder of the flat surfaces which bound the water and steam spaces. Into all these different strains it will be necessary for us to inquire, and we shall now proceed to illustrate each kind by example.

We will first take the case of an ordinary Cornish boiler with a single flue.

Assuming the boiler to be single riveted throughout its length, the strength of the joint will be just half that of the solid plate; but if it be double riveted, the strength will be seven-tenths of that of the solid plate. Hence, for single riveted boilers we have the following rule:—

RULE.—*To find strain per sectional square inch in a boiler, multiply the pressure of steam per square inch in the boiler by the radius of boiler in inches, and divide by the thickness of the plate in inches.*

Let pressure of steam be 60 lb. per square inch. Radius of boiler 30 inches, thickness of shell $\frac{1}{2}$ inch.

60 lb. per square inch.
30 inches radius of boiler.

Thickness of plate ·5)1800·0

3600 lbs. tension per sectional square inch.

The safe working strength of solid plate in tension is

The fact that the length of the tube comes into the calculation in considering crushing strain on wrought iron tubes, but not in the case of bursting strain, prevents us from making any general comparison as to the relative strengths of tubes under the two sets of circumstances.

We will next consider the strengths of stayed surfaces of a flat form; they evidently assume the conditions of solid continuous beams uniformly loaded, and, therefore, may be treated as follows:—

Rule.—*To find the requisite thickness for a stayed surface in inches, multiply the square root of the steam pressure in lbs. per square inch by the greatest distance between the stays in inches, and by* ·008.

Let the pressure be 60 lbs., and the distance between stays 7 inches.

```
    7)60(7 7 square root of pressure (nearly).
      49
   147)1100
       1029
         71
         ..
```

```
   7·7 square root of pressure.
     7 inches distance of stays.
  53·9
  ·008
 ·4312 inches thickness of plate.
```

Practically this would be ½ inch.

The nature of the strains on the stays connecting the

flat sides will be simply tensile, and the diameter of the stay will be found as follows :—

Rule.—*To find the diameter of stay in inches, multiply the distance between the vertical rows of stays by that between the horizontal rows, multiply the product by the pressure in lbs. per square inch, and divide the square root of the product by* 70.

Let the vertical and horizontal distances be respectively 6 inches and 8 inches, and the pressure of steam per square inch 60 lbs.

```
            6 inches vertical diameter.
            8   „   horizontal   „
          ----
           48
            60 lbs. per square inch.
          ----
          2880 product.
          ====

          5)2880(53·6 square root nearly.
            25
          ----
       103) 380
            309
           -----
      1067) 7100
            6402
           -----
            ....
           =====

Constant 70)53·6
            -----
             ·765 inch diameter of stay.
```

Practically the stays would be $\frac{7}{8}$ inch diameter.

With regard to spheres and spherical, or egg ends, it

may be observed that they are twice as strong as cylinders of like diameter and thickness, when exposed to internal or bursting strain.

The flat ends of cylindrical boilers are usually made of stouter plate than the barrel, but they are also tied in with stays, or else with triangular pieces of plate fixed with angle irons, like the gusset plates of a girder.

The horse-power of a boiler is ascertained from its heating surface, one horse-power being represented by 8·5 square feet of horizontal surface, or 17 feet of vertical heating surface; and in the furnace, by from ½ to ⅔ of a square foot of grate surface.

Of the fittings necessary for boilers, the safety valve is one of the most important, if not first in importance. It should be made of sufficient size to permit the escape of the steam as rapidly as the boiler is capable of generating it, for otherwise it is evident that when an engine is suddenly stopped and left standing, the steam which continues to form accumulates in the boiler, and the pressure rapidly increases, perhaps to a degree inconsistent with safety.

As a rough practical rule, we may assume that the safety valves should afford as much area for the escape of steam, as is to be found in the steam passages supplying the engine which the boiler is arranged to work, but this is assuming that those passages are correctly designed, which, however, may not be the case; wherefore we give a special case. The general rule for the flow of gases through orifices is as follows:—

Rule.—*To find the quantity discharged in cubic feet per minute, divide the pressure per square inch in lbs. on the gas by its density as compared with water, multiply the square root of the quotient by the area of the orifice in square feet and by 474.*

From this general rule a special one is deduced for the special case under consideration.

RULE.—*To find the diameter of the safety valve in inches, divide the square root of the quantity of steam generated in cubic feet per minute by ten.*

Let the boiler be capable of supplying 950 cubic feet of steam per minute,—

3)950(30
9
60)50

10)30
3 inches diameter of safety valve.

It will be necessary to explain here why the pressure of the steam in the boiler is not taken into consideration in this rule. The quantity of steam evaporated from a given bulk of water varies inversely as the pressure at which it is working, its density or weight varying at the pressure; but the pressure also increasing with the density, supplies the greater force requisite to project the denser gas as quickly as the lighter.

Boyle and Marriott's law is, that the density of a gas is as its pressure, and inversely as its volume; but this is not exactly true of steam, because the temperature of the steam is not constant, and it is not a permanent gas; but for all practical purposes the above rule is sufficiently correct, as the constants are assumed to allow a sufficient margin.

The next point to be considered is the loading of the valve. The load may be applied direct by spring or

weight, or through the intervention of a lever, the latter being the most usual method to avoid having very cumbrous weights. When the weights are applied direct to the valve, and hang inside the boiler, allowance must be made for their being immersed in water, as iron weighs one-seventh less in water than in air; so, if we want a load of six pounds effective, and the weights are to be immersed in water, the actual weight used must be seven pounds. If, however, the weights be of lead, they will lose one-eleventh; therefore for every ten pounds effective weight we must, in this case, have eleven pounds actual weight.

The direct load required on a safety valve may be calculated in the following manner:—

Rule.—*To find the direct load requisite for a given safety valve, square the diameter of the valve in inches, and multiply by the pressure of steam in lbs. per square inch at which the valve is intended to open, and by* ·785; *the product will be the required load in lbs.*

Let the diameter of the valve be four inches, and the extreme pressure to be maintained in the boiler forty lbs. per square inch,—

```
       4 inches, diameter of valve
       4    "       "       "
      ---
      16 square of  "       "
      40 lbs. pressure per square inch.
     ---
     640
    ·785
    ----
    3200
   5120
  4480
  -------
  502·400 lbs. load on valve.
```

To find the exact amount of weights requisite, we must deduct the weight of the valve itself from this gross weight. We will assume the weight of valve and rod at 5 lbs., which, deducted from 502 (dropping the ·4), leaves 497 lbs. extra weight to be put on the valve. If the weights are to be iron, and suspended in water, we must add $\frac{1}{6}$ to the above.

497 lbs. weight required in air
 83 lbs. $\frac{1}{6}$ ditto (nearly)
580 lbs. weight required if in water.

If the load is to be applied through the intervention of a lever, the amount of weight hung to the lever will thus be found :—

RULE.—*To find the weight at the end of the lever to produce a given load in lbs. on the valve, multiply the load by the distance in inches from the fulcrum of the lever to the centre of the valve, and divide the product by the distance in inches from the fulcrum of the lever to the point of suspension of the weight.*

Let the load required on valve be 497 lbs.; distance from fulcrum of lever to centre of valve 4 inches; and distance from fulcrum of lever to weight 24 inches.

```
                              497 lbs. load
                                4 ins. dist. fulcrum to valve
Dist. fulcrum to weight 24)1988(82·8 (say 83 lbs. at end
                           192             of valve lever.
                            68
                            48
                            200
                            192
                              8
```

Of pressure gauges we have not much to say, except that, for ourselves, we put little faith in any but mercurial gauges, which, however, in the simple form, cannot well be applied for high pressures, on account of the great height of the column of mercury requisite to balance a high pressure; and of course it is impracticable to apply them to any but stationary boilers.

In making and repairing boilers of all descriptions caulking is to be avoided as much as possible, tight joints being better secured by good riveting—that is, by riveting the plates carefully, so that the contraction of the rivets in cooling will draw the plates into close contact. Any slight interstices will soon rust up.

In order to make good close joints, the plates should be brought into the closest possible contact before riveting, so as to prevent the rivet forming a collar between the plates through which it passes, and then the contraction of the rivets will do the rest.

We will now pass on to make some remarks on the subject of tanks, though this work is so similar to boiler work, that but few observations will be necessary.

An ordinary tank, open to the atmosphere, will obviously have no other strain to withstand beyond that which is due to the weight of water in it; hence the maximum strain on the plates will be reached when the tank is full of water. The tanks made of wrought-iron plates should, if of any considerable depth, be strengthened at intervals by angle or tee-irons, in order to prevent the sides from being bulged out; and this being the case, the proper thickness of plates at any depth can be found from the following rule:—

RULE.—*To find the thickness in inches of any plate necessary to resist the pressure of the water, divide the*

distance in feet between the strengthening ribs by 20, and multiply the quotient by the square root of the depth in feet from the surface of the water to the plate under consideration.

For example, let a tank be 10 feet deep; the distance between the ribs 3 feet; then the depth of the bottom plates will be 10 feet: hence, as 3·16 is the square root of 10, we proceed thus:—

```
20)·3   feet, distance between ribs
  0·15
  3·16 square root of depth in feet
    90
   15
  45
·4740 inches, thickness of bottom plates.
```

The thickness of the bottom plates should not be less than ½ inch.

Having settled the thickness of the bottom plates, it becomes the next task to determine that of the side plates. The thickness must be calculated for the depths of the lower edges of each tier of plates where the maximum strain will come upon them.

Such a tank would be made with three tiers of plates to form the sides, each tier being 3ft. 4in. (3·34 ft.) deep, so as to make up the total depth (10ft.) of the tank, and upon these depths we can act.

The bottom tier of plates should be of the same thickness as the bottom plates of the tank, that is ½ inch, as their lower edges are the same distance below the surface of the water as the bottom plates.

The lower edge of the second or middle tier of plates will be 6ft. 8in. (6·67ft.) below the top of the tank. The square root of 6·67 is 2·58.

```
20)3     ft. distance between ribs
   0·15
   2·58  square root of depth in feet
    120
     75
   30
  ·3870  inches thickness of middle air.
```

These plates, therefore, would be made $\frac{7}{16}$ inch in thickness.

The lower edge of the top row of plates will be 3ft. 4in. (3·34 ft.) below the top of the tank. The square root of 3·34 is 1·82.

```
20)3     ft. distance below ribs
   0·15
   1·82  square root of depth in feet
     30
   120
   15
  ·2730  inches thickness of top row.
```

So the top row of plates should be $\frac{5}{16}$ inch in thickness.

The angle irons, tee-irons and cover strips should be of scantling proportionate to the thicknesses of the plates. The ½ plate should be fastened with ¾ rivets, the cover strips being 4 inches wide; the angle iron for joining the bottom of the tank to the sides should be 2½ inches by 2½ by ½ inch thick; thus allowing 2 inches width of plate for each rivet which may be 2 inches pitch. Vertical tee-irons [illegible] *inches by 3* inches by ½ inch thick.

The $\frac{5}{16}$ plates should have $\frac{5}{8}$ rivets, cover strips $3\frac{1}{2}$ inches wide, and stiffening tee-irons 4 inches by 3 inches by $\frac{1}{2}$ inch thick.

The $\frac{5}{16}$ plates to be jointed the same as the $\frac{7}{16}$ plates. Across the top of the tank should be carried stiffeners formed by bending the vertical stiffeners, so as to form frames, and the edges of the tank may be stiffened by angle irons 8 inches by 3 inches by $\frac{3}{8}$ inch thick.

These calculations refer to a tank which is always open to the atmosphere, inasmuch as no pressure can be accumulated within it other than that due to the weight of water which it contains. But there are some tanks which may become subject to internal pressure from the accumulation of steam,—such are feed water tanks placed around the uptake of a boiler, in which the temperature may rise sufficiently high to cause the generation of steam, accompanied by increased pressure, which, if not allowed to escape, is very likely to cause an explosion, such as that which occurred on board the Great Eastern a few years since, when one of the feed water tanks burst through the accumulation of steam.

We will now proceed to consider the manufacture of girders, but shall not enter into the calculations of strains on the main elements, as that would be foreign to the object of the present volume, besides entering into matters which would require a treatise to themselves.

It is of course always presumed that the working drawings sent into an iron yard are *really* working drawings; but experience shows that such is frequently not the case, wherefore it behoves superintendents to carefully examine all drawings previously to commencing the manufacture, in order to ascertain if they be practically detailed.

Not unfrequently it will occur that tee and angle irons are by thoughtless and inexperienced designers, shown of such sections as will not allow room for the rivets required to retain them in position: thus, we have seen a tee-iron shown $2\frac{1}{2}$ inches by $2\frac{1}{2}$ inches by $\frac{3}{8}$ inch shown on drawings, and specified to be fastened by $\frac{3}{4}$ inch rivets. A tee-iron $2\frac{1}{2}$ inches on the back and $\frac{3}{8}$ thick will have $2\frac{1}{8}$ inches clear, of which one-half or $1\frac{1}{16}$ inch will be on each side of the web; and if a $\frac{3}{4}$ inch hole were taken out of this, it would only leave $\frac{5}{32}$ inch metal outside the hole; and, in short, such a hole could not be punched without bursting the metal; and, in fact, the very width to carry a $\frac{3}{4}$ rivet should be $1\frac{1}{2}$ inches clear on each side of the web, and this is hardly sufficient to be quite satisfactory.

The strength of plated work depends very materially upon the mode in which the rivet holes are formed; that is to say, whether drilled or punched, and truly pitched or otherwise. Practically, a hole cannot be drilled in a thin plate with such accuracy as it can be punched, for as soon as the point of the drill is through the plate it loses its guidance, and unless the drill is very stout it will spring and wobble about, and so produce a hole that is far from being circular. The objections that have been raised to punched holes is that they are always conical, and that punching greatly strains and injures the plate. It must certainly be admitted that in ordinary punches, where the punches used taper very much, these assertions possess a considerable claim on our notice, though they may be much reduced in importance by adopting a perfectly cylindrical punch made hollow, or slightly sunk at the bottom in order to form a cutting edge, so that the bars of iron removed are cut out, not burst out as they are by the inferior punches.

By combining the two processes, punching first, and

then running a drill through the punched holes, very satisfactory results may be arrived at; thus, if a $\frac{3}{4}$ hole is required, it may be punched $\frac{5}{8}$ inch, and drilled out to $\frac{3}{4}$ inch diameter; thus taking out any conicality that might exist in the hole, and smoothing its interior surface as well. This in our opinion is a decided improvement on drilling out of the solid, except where very thick plates are used; and plates should certainly not be drilled in cases where the diameter of the hole exceeds the thickness of the plate.

If the holes are not truly pitched and accurately made, they will not come opposite one another when the work is put together, and so they have to be rhymered out, which enlarges them above the required size, or else they must be "wrenched" into positions by means of drifts or "podgers;" thus straining the plate and rendering the work inaccurate when done. The proper pitching of the holes is really a matter of the greatest importance, for when the plates are riveted together, there should be no strain upon them until sustaining the load for which they were designed.

It is impossible to get the pitch exactly true unless self-acting punching machines be used; but it is worthy of note how very few of the large bridge manufacturers use them, as notwithstanding that although their original cost is something considerable, they soon save it by economising the labour commonly expended in rhymering and drifting out the holes, besides turning the work out quicker and far better than the ordinary machinery. It would be better if engineers generally would rather insist on thoroughly good punched work, instead of specifying, as is so constantly done, that all holes shall be drilled out of the solid, a clause which is often subsequently cancelled.

In order to execute the work satisfactorily and with economy, hand labour should as far as possible be replaced by machinery; but this cannot in all cases be done, as machinery only pays when there is a large quantity of work of similar description to be executed; thus, if we have a few angle irons to bend for knees to a small girder, it would not be worth while making tools to bend them, but if there be a great quantity of the same size and form there is a great saving in pressing them, over smithing the ends in the ordinary way; and, in addition to this, the pressed knees will be far more uniform than those turned by hand; which is a great advantage in some cases, such as where bridge-work is shipped away from the iron yard piece-meal, under which conditions it is very desirable that knees should be, if possible, interchangeable,—any one knee fitting any part of the girder,—thus saving much tedious work in marking. It not often that such a mode of shipment is adopted, but we have known of some recent cases in which colonial girders, after being put together in England, have been taken entirely to pieces and so sent out without one rivet in the work from one end to the other, and this even in bridges of 30 feet span, which might be shipped whole.

In smithing the bars of tee and angle iron knees, it is usual to draw the end to be bent down over a cast-iron block, and it may be observed that the bar by so doing is lengthened, being thinned down on the table or back, and thus weakened; whereas when the knee is bent by being pressed between dies, the metal is rather thickened at the bend, thus increasing the strength just where it is required, that is, at the angle, and making no waste; for the iron for making these knees would probably be ordered long enough *to make the knees* without relying upon the extension under

the smith's hands; hence, all the extension that takes place will have to be cut off and go to waste.

In pressing these knees into shape, an hydrostatic press, worked with an accumulator, is most useful, as with it is obtained great force, together with tolerable rapidity of action. The dies are made of cast-iron, and attached to the table of the press, and to its crown-plate. A press, with a ram of about 10 inches diameter, working under a pressure of 1,400 lbs. per square inch, will be found sufficient for most work of the class to which the foregoing remarks apply.

The same press may also be used for making joint-plates, where a large number of the same size are required, in preference to shearing them to shape in the ordinary shearing machine.

There is another plan of bending angle and tee-iron knees by hand, not yet referred to, which consists in cutting out a V-shaped piece at the point where the elbow is to be made, the cut being in the web, then bending the bar, and so bringing together the cut edges of the web and welding them up; but here is introduced the danger of getting an unsound shut, and that just at the point where the greatest strength is required, as the object of a knee or elbow is to preserve some definite angle, and therefore the strain upon it is at the bend.

Wherever a joint occurs *in tension* in work, half-rivet holes at the ends of the plates must be carefully avoided, as a rivet half in one plate and half in the next is useless, and might as well be left out, for the two plates, if pulled asunder, would receive no hindrance from such a rivet; but, when the strain is in compression, it is a matter of no moment, as the plates are pushing towards the rivet, not pulling away from it.

In order that those who carry out the plating of girders should be acquainted with the *kind* of strain that comes upon the various parts of the job, we subjoin some criteria of strain for the guidance of such as may not have turned their attention to the matter:—

1. *In an ordinary arch, the strain is compressive throughout.*

2. *In an ordinary suspension chain, the strain is tensile throughout.*

3. *In a bowstring the top member is in compression.*

4. ,, ,, ,, *bottom member or tie is in tension.*

5. ,, ,, ,, *bracing bars are alternately in compression and tension.*

6. *In a plate girder the top member is in compression.*

7. ,, ,, ,, *bottom member is in tension.*

8. ,, ,, ,, *web is under shearing or vertical strain.*

9. *In a lattice girder the top member is in compression.*

10. ,, ,, ,, *bottom member is in tension.*

11. ,, ,, *all bars with the top nearer to the centre line than the bottom are in compression.*

12. ,, ,, *all bars with the bottom nearer to the centre line than the top are in tension.*

In putting girders together, under all circumstances, a certain amount of camber or upward curving should be given them in order that, when under their load, they may not deflect below a straight line. The amount of camber given is usually 1 inch for every 40 feet of span: this camber may most readily be given by making the top flange somewhat longer than the bottom flange, and building *on blocks* suited to the intended curvature of the girder.

Before concluding this chapter, it is necessary to speak of machine riveting as compared with handwork.

Although there has been a prejudice against machine riveting, yet, at the present time, it is found practically to be decidedly the better of the two, as it makes closer work, and generally with less danger to the rivet-head, while the head being made at one stroke, the girder escapes the jar and vibration due to the repeated and incessant impact of the flogging hammers used in the hand process.

CHAPTER X.

FITTING, TURNING, AND ERECTING.

The various parts of a machine having been made in detail, it remains to fit them together, and mark them, so that their positions may be known at any future time when it may be necessary to take the machinery to pieces, and for repair or removal. In the present chapter we purpose entering upon the subjects of fitting, turning, and erecting, assuming, however, that in all cases the masonry foundations are put down independently of the maker of the machine.

Each of the separate parts of a machine having been wrought as near to the exact size as may be determined by gauges and measuring instruments, it is then applied to that portion to which it has to be attached, and chipped, filed, or scraped, as may be necessary, until the desired fit is obtained.

Standards intended to be carried on a bed-plate are generally cast with fillets or chipping-pieces, by which the level of the frame may be adjusted without incurring the expense of planing the whole of the foot or bottom of the standard.

Before placing the standards in position on the bed-plate, the centre lines should be carefully laid down and *marked with a* scriber or centre-punch; and if there be

two standards, each one should be marked, and a corresponding mark be made on that part of the bed-plate to which it belongs.

The standards having been fitted and temporarily bolted to the bed-plate, the next step is to bore out the bearings for the shafts, if there be any, and this ought to be done with the standards in position, so that the two bearings of any one shaft may be bored out by one boring bar passing through such bearings to ensure accuracy of the axes, as otherwise the centre line of one bearing might not be lineable with that of the bearing destined to receive the other end of the shaft, under which circumstances it would be simply impossible to obtain a true fit; and if the bearings were scraped out of truth to enable the shaft to revolve, the result would be loss of bearing surface and undue friction in the bearings.

When the bearings, whether of brass or iron have been bored, the shaft with the journals turned upon it should be tried into the bearings in the usual way, and the latter, if necessary, scraped until a correct fit is attained.

If it be impracticable to bore the bearings as described above, or even very inconvenient, the bearings may be bored separately, but in that case, very great care and superior workmanship are indispensable.

In such a case as fitting bearings in plummer blocks to carry the main shaft of an engine, the sides of the frames upon which the plummer blocks are to rest may first be planed at one operation; that is, the cut of the planing tool must be carried right through across the two sides of the frame, and the bottoms of the plummer blocks should also be planed. The holes in the frame to be left out until the exact positions of the plummer blocks are ascertained by trial. The plummer blocks with the bearings in them are

then to be bolted to the slide rest of the boring lathe, and both bored off the same bar and with the same cutters; this will ensure accuracy of centres, as far as the vertical measurements are concerned; laterally the positions must be determined by centre lines on the bed-plate or framing of the engine, any slight adjustment that may be necessary being made by means of wedges acting between the ends of the plummer blocks and snugs cast on the frame.

The exact position of the plummer blocks having been ascertained, the holes in the bed-plates may be drilled for the bolts by which the plummer blocks are intended to be secured to it, so that if removed at any future time there will be no difficulty in at once replacing them accurately, without the necessity of a fresh adjustment.

It may be said, and justly so, that with accurate workmanship and correct working drawings, there should be no need for so much opportunity for adjustment, but, unfortunately, that accurate workmanship is not always available, and our attention is more especially drawn to possible or probable cases of difficulty, than to those where every facility is at hand; and it is most important that the student should not confine his attention merely to successful works, carried out in the best manner by the most competent mechanics, but he should also especially study the errors of others, so that he may not fall into like ones himself; and should also qualify himself to deal with any emergencies which may be likely to arise; and one of the most trying presents itself when called upon to construct a machine without having skilled or intelligent artisans to execute the designs; for unless the designer is in a position to devise modes of effecting the manipulations necessary to the completion of his work, he must in preparing his plans take into consideration the capabilities of

his workmen; and may, perhaps, be compelled to abolish an elegant mechanical arrangement and substitute it by some rougher and less efficient contrivance, on account of the inaptitude of those who are to work to his instructions.

The principle of ultimately determining positions in erecting machinery has, in at least one instance, been carried so far that every engine made of which a similar one had not previously been constructed, was made from drawings with little more than the centre lines upon them, the work being developed in the shop, and the complete working drawings made after the satisfactory completion of the machine, and therefore exactly representing the engine as executed.

Of the great value of such plans there cannot be the slightest doubt, and more especially when they have reference to machinery going abroad, as in case of any renewals being required, they can be made from the drawings with the certainty of fitting when sent out to their destination.

To return to the details of the fitting shop, let us take the fitting together of certain parts of an ordinary steam engine, in order to describe the manner in which the work should be executed.

Beginning with the cylinder and its appurtenances, the mode of packing the piston may first be considered.

The old-fashioned method, of using one or two wide spring rings for pistons of ordinary size, has been almost superseded by the system introduced by Mr. Ramsbottom of using small rings, varying in width from a quarter of an inch upwards, and each one being placed in a groove by itself. The periphery of the ring is not exactly cylindrical, the edges being rounded off with a view to obviate

the possibility of cutting the interior of the cylinder with the edges of the packing rings. This mode of packing is very successful, but some makers fall into the error of putting a steady pin into the body of the piston where each ring is split, in order to prevent the ring from turning round; this, however, is quite unnecessary, as if the ring did turn it would not matter, even (which is highly improbable) if the split part of two rings did come in the same line, as the amount of leakage would be insensible, and moreover, as the rings wear down, and the cylinder wears larger, there will be a considerable interstice where a portion of the ring is cut away to make room for the insertion of the steady pin.

In boring out the cylinder, which is done with a boring bar and boring head, the latter furnished with recesses or slots in its periphery for carrying cutting tools, the finishing cut should be taken from end to end without stopping the tool, except for a minute or two for adjustment, because as the cylinder heats and expands under the action of the cutting tool, it will, when the machine is stopped, cool and contract, the result of which would be equivalent to giving the tool a deeper cut at some portion of the length of the cylinder, thereby producing a ridge which would be detrimental to the action of the machine when at work. The boring bar should be used with at least two cutters diametrically opposite to each other, for if but one were used, the vibration of the boring bar as the tool passes over places of various degrees of hardness, will tend to vitiate accuracy of the cylinder; three cutting tools placed at equal distances round the boring head, that is, making angles of 120 degrees with each other, will be better than the two, but it is very questionable whether any gain is experienced in increasing the number of cutters

beyond this, except in saving time by taking several cuts at once, which in large cylinders is of importance.

The final cut in boring a cylinder should be taken with a point tool which yields the most accurate work on the mean surface, but as a matter of course traces as it were a very fine spiral groove upon the interior of the cylinder, this, however, is immaterial, as in the course of two or three days' working it is worn away, and the surface of the cylinder becomes bright. If the slight roughness of the cylinder be removed by any kind of grinding or polishing, two possible evils are unnecessarily called into existence, one is the danger of interfering with the circular contour of the cylinder, the other that of some portions of the polishing powder becoming imbedded in the substance of the cylinder, and subsequently acting to wear away the piston when the engine is at work.

By leaving the work in its truest state, though rough, —that is, as it comes from the boring lathe,—the most satisfactory results are attained; for if the metal be fairly homogeneous, it will wear away equally, and although wearing, will remain true.

In fitting the valves to the port faces of the cylinder, it used to be the rule, after getting them as true as possible with the planing machine and file, to face them up with a scraper to a surface-plate, also, in like manner, facing up the ports; but there are disadvantages in having surfaces which are moving upon each other too nearly approaching to accuracy, for they will sometimes bind and stick so firmly as to necessitate the removal of the slide-jacket in order to release the slide.

Under these circumstances it is obviously advisable to omit the process of facing with the scraper; and, having

made the port and slide faces true, to let them wear each other smooth. These remarks, as a matter of course, apply equally well to expansion-valves and others of a sliding character.

In sliding parts, such as those above referred to, there is a constant tendency to bind, produced by the steam pressure on the back of the valve; but no such tendency is in action on certain other sliding surfaces, such as guides, which may be faced up if desired. The method of using the surface-plate or, more properly, the planometer, is as follows:

The planometer is a cast-iron plate, made with one surface,—a perfectly true plane, so far as it is possible by the best means and most careful manipulation to make it so,—and, on the under side, the plate is furnished with numerous strong ribs to obviate the possibility of its losing its truth by deflection.

The work to be surfaced is first planed or chipped, and filed up to a true but rough plane,—that is to say, the mean between the tops of the prominences and the bottoms of the intervening depressions is a true plane,—and the object now is to remove these prominences, and arrive at a surface which shall be smooth and a true plane: the process is as follows: A small portion of "ruddle" or other colouring matter is rubbed on to the surface-plate, to which the work is then applied (unless it be very heavy, in which case the surface-plate is applied to the work, being furnished with handles by which it may be readily moved), and gently moved about upon it.

It follows that, as the work will only touch the surface-plate with its highest prominences, those points will be marked by the red colouring matter which had been rubbed upon it; the points so marked are levelled down

with a scraper, which may be made by grinding up the end of an old three-square file: the work is then again applied to the surface-plate, and moved about upon it, when more points will be found to have become coloured; and this process is continued until, on applying the work to the surface-plate, it becomes generally coloured with the ruddle, when the surface may be regarded as representing a true plane; its appearance, when the colour is cleared off, will be marbled or mottled.

This process of surfacing is necessary in making many machine tools requiring great accuracy of workmanship, such as the beds and slides of lathes, &c.

In scraping up cylindrical surfaces, such, for instance, as the bearings of steam-engine shafts, the process is similar; but a cylindrical surface is used in place of the planometer.

The journals of the shaft being accurately turned and finished, and the brasses bored out as true as possible, the ruddle is rubbed on the journal of the shaft, which is then applied to the bearings, which are scraped until they become the counterpart of the cylindrical surfaces of the journals: they may not be true cylinders, but as there is no lateral movement, all that is absolutely necessary is to ensure that at every point in the length they are truly circular.

It may be inquired why the turned journal is selected as the criterion of accuracy in this case, hence it will be as well to explain.

Generally, speaking of small diameters, any piece of work can be more accurately turned than bored, and for this reason, the turned work is cut by a tool having no neck, and which may be made as rigid as we choose;

whereas to bore a piece of work of small diameter, it is necessary to use a side tool having a neck as long as the work itself; hence the liability to spring or chatter, and the resulting inferiority of the work. Again, in turned work, the file can be applied to smoothe it while it is running in the lathe; but this cannot be done with bored work with any chance of success, as the file would have to be held longitudinally in the work, and by one hand only, so that it could not be maintained in a true position; hence it will be seen that turned work, as a rule, is more accurate than that which is bored by a tool fixed on a rest, though if there is room to use a boring bar, there is no reason why boring should not be accomplished as well as turning.

In speaking of true work, of course the term must be taken with a reservation as meaning *as true as it is possible to get it*, there being no such thing as absolutely true work.

Probably the most accurate fit ever known was that of the piston of a steam-engine indicator, made as usual, solid, which would not, when dry, admit of being passed into the corresponding cylinder, although, when lubricated with fine oil, it could be introduced therein.

In making all elements of machinery, regard should be had to the direction in which they move, either in bearings, through eyes, packings on guides, or otherwise, and the last tool applied to them should move in the same direction as they are intended to, so that, if there be any marks, they will be in the line of motion, and not across it, which might materially accelerate the wear of the parts. In revolving shafts, the last cutting action should be round them, in the same direction as the tool acts in the lathe *hence, if* they are filed in the lathe to smoothe the journals,

the file should only be moved very slowly laterally, so that any marks would be in the form of a screw of such fine pitch as practically to represent circles.

In like manner, in scraping up the bearings for shafts, the scraper should not be moved in the direction of the length of the bearing, but round its surface transversely.

If we consider a piston-rod, the case is different, as its ultimate tool-motion through the stuffing-box is longitudinal; hence, the marks should also be longitudinal; but the rod being turned in a lathe, when it is reduced to its proper size the tool-marks will be round its periphery, wherefore it is necessary it should be draw-filed.

The process of draw-filing consists in placing the file across the work in the usual manner; but instead of traversing it in the direction of its (the file's) length, it is moved laterally, so as to act longitudinally upon the work in the lathe, which may be kept slowly revolving in order that the work may be equally acted on all over its surface.

By this process, while the file (being moved sideways) does not make any decided marks upon the work; yet it suffices to remove any circular marks which previously existed.

Not only for piston-rods is this required; but for plunger pumps, and more especially for such as are of large diameter, in which any circular marks would materially affect the wear of the packing and so produce an unwonted leakage, a leakage which becomes of great importance in pumps of two and three feet in diameter, and having a stroke of eight or ten feet.

A few words may not here be out of place as to the relative uses of point and spring tools for turning.

The general course adopted for ordinary work is first to

turn the rod true with the point tool, and then to finish it with the spring tool, the latter being used with only a very light cut. The spring tool, yielding to any hard spot in the material instead of forcing through it or tearing it out, leaves a cleaner finish than the point tool, although the work is not so accurate; hence, when great truth is required, the spring tool should not be used.

CHAPTER XI.

SCREW CUTTING.

There are three distinct methods used for the production of screw: 1st, by means of taps and dies; 2nd, by chasing in a lathe; and 3rd, by cutting in the lathe with a single-pointed tool.

Screw threads are of two forms, V-shaped and square. The former may be produced by any of the three methods mentioned above; but the latter are invariably cut in the lathe.

The square-threaded screw is most exact in its working; but the V-thread is obviously the strongest in proportion to the length of screw, inasmuch as at the bottom of the thread none of the metal is cut away in the latter, whereas in the former kind of screw the space between the threads being equal to the thickness of the thread, it follows that one half of the material is removed in cutting the screw. With the V-thread there is, however, a tendency to force outwards or burst the nut which does not exist in the square-thread, as the working surfaces of the latter producing motion in the direction of the length or axis of the screw lie at right angles to such axis, and therefore produce no lateral wedging action. We will now

inquire into the strength of such screws relatively to the strength of the solid cylinder upon which they are cut.

The strength of the solid cylinder in tension will vary as the square of the diameter ; but the strain on the screw thread is a shearing force tending to "strip" the screw ; hence there will be some constant relation of length of thread to the diameter of the screw, which will in all cases give equal strength in the body and the thread.

We will commence with the V-thread. It is hardly to be inferred that practically the base of the thread actually occupies the whole surface of the cylinder carrying such thread, as that would involve the necessity of having an exactly true angle, whereas the bottom of the thread is always rounded to some extent; hence it would not be prudent to calculate upon more than three-fourths of the surface of the cylinder as area to resist shearing strain.

The strength of the body of the screw to resist tensile force may be thus found :—

RULE *(a).—To find the safe working tensile strain on the screw, multiply the square of the diameter of the screw in inches (measured at the bottom of the thread) by* 4. *The result will be the safe load in tons.*

Let the screw measure two inches in diameter at the bottom of the thread.

2	inches	diameter	of	screw
2	"	"		"
4	square of			"
4	constant			
16	tons safe load.			

To find the shearing surface of the screw in square inches *we have the following* :—

Rule.—*To find the shearing area of a V-thread screw, multiply the diameter (measured at the bottom of the thread) in inches by the length of screw in inches and by 2·35. The product will be the area in square inches resisting the shearing force.*

Let a screw be 1½ inches in diameter at the bottom of the thread, and 2 inches in length.

1·5	inches diameter of screw.
2	„ length „
3·0	
2·35	constant
150	
90	
60	
7·050	square inches of shearing area.

Allowing four tons per square inch as safe working strain, the strength of this screw would be

7·05	square inches
4	tons per square inch
28·20	tons safe load.

What is required, however, is to have the screw of such a length that it may be equally strong with the bolt or body; hence, the diameter being given, the following rule will apply:

Rule (*b*).—*The diameter of a screw being given, and also the load it is proposed to work under, to find the correct length of the screw, divide the load in tons by the diameter of screw in inches and by 9·4; the quotient will be the length in inches.*

Taking the dimensions and load from the first example, we thus find the proper length of screw, the diameter being 2 inches, and the safe load 16 tons.

Diameter of screw 2)16 tons safe load

```
9·4)8·00(0·85 inches (nearly).
    752
    ---
     480
     470
     ---
      10
```

To find the ratio of length to diameter, the latter must be divided by the former.

```
·85)2·00(2·35 (nearly).
    170
    ---
     300
     255
     ---
      450
      425
      ---
       25
```

In order to ascertain if this ratio holds in other cases, it is desirable to take another size of screw, and work it out by the two rules in like manner; let the screw be three inches in diameter.

First find the safe load on the body of the screw by rule *(a)*.

```
 3 inches diameter of screw.
 3   "       "        "
--
 9 square of "        "
 4 constant
--
36 tons safe load.
```

This gives the necessary data wherefrom to determine the length of screw-thread proper for this diameter by means of the second rule *(b).*

```
Diameter of screw  3)36 tons safe load.
                 9·4)12·0(1·276 inches (nearly).
                     9·4
                     ---
                     260
                     188
                     ---
                      720
                      658
                      ---
                       620
                       564
                       ---
                        56
```

From which is found the ratio

```
1·276)3·000(2·35 (nearly).
      2552
      ----
       4480
       3828
       ----
        6520
        6380
        ----
         140
```

Showing that the same ratio between diameter and length of screw obtains in each case. It is desirable to find now what proportion is this length to the diameter of the screw.

```
2·35)1000(0·425 (nearly).
     940
     ---
      600
      470
      ---
      1300
      1175
      ----
       125
```

From this it will be seen that, to keep equal strength in the body of the screw and the thread, the length of the screw should be a little less than half the diameter of the bolt.

The common custom in making bolts and nuts is to make the depth of the nut, that is, the virtual length of the bolt, equal to the diameter of the bolt, or rather more than twice as strong as is necessary; the object of this is to allow for bad workmanship in the screw-thread; but, strictly speaking, bad workmanship should not be suffered, for, if it be once admitted as a matter to be taken into consideration, then calculations become almost useless; for it is impossible to say how far bad work will vitiate the strength of the various details of machinery; therefore we maintain that the work should be carefully and properly designed, and good work should be *insisted on* by the superintending engineer, and all defective work at once and unhesitatingly rejected. Such a course would be better, both for the manufacturers and the buyers of machinery; for, with good work, machinery may be made much lighter than when allowances have to be made for defective manipulation of materials.

We will now proceed to deal with the square-threaded screw, merely, however, observing that we may practically in the foregoing case assume 0·5 instead of 0·425 as the ratio for the V-thread, as it is simpler and sufficiently near for all purposes; so that, for the V-thread, the length of screw or nut should not be less than half the diameter of the bolt or body of the screw.

In the square-threaded screw, one half of the shearing surface is cut away in cutting the thread, so we have but half of the whole surface of the body of the screw upon which to rely to resist the shearing strain brought upon

it ; but, in regard to tensile strength, the resistance is the same as with the V-threaded screw.

In order to determine the tensile strength of the body of the screws, the rule *(a)* may be applied.

Let the screw at the bottom of the thread be 2½ inches in diameter.

```
  2·5 inches diameter of screw.
  2·5    "        "        "
  ---
  125
  50
  ----
  6·25 square of diameter of screw.
     4
 -----
 25·00 tons safe load.
```

To find the necessary length of screw, we have the following rule :—

RULE.—*To find the necessary length in inches of a square threaded screw to afford equal strength with the body of the screw, divide the working load in tons on the screw by its diameter in inches, and by 6·3, the quotient will be the required length in inches.*

Acting upon the foregoing data, we find as follows : —

```
Drain of screw  2·5)25·0(10
                    25
                    --
                     0
                    ==

          6·3)10·0(1·587 inches (nearly).
               6·3
              ----
               370
               315
               ---
                550
                504
                ---
                 460
                 441
                 ---
                  19
```

From these figures the ratio may now be obtained, and it will be shortest to take the ratio of length to diameter, as it will be unnecessary to repeat the test as to the same proportions holding good for different sizes, as that being true of the V-threaded screw, must also be true of the square-threaded screw. The diameter is 2·5 inches, the length 1·587 inches.

```
2·5)1·587(0·634 (nearly).
    1·50
    ----
       87
       75
      ---
      120
      100
      ---
       20
       ..
```

Whence we find that the length of the square-threaded screw should be a little more than half its diameter, in fact, nearly two-thirds, but for simplicity we may call the multiplier 0·7. This shows the strength of the V-thread to that of the square thread, for equal lengths of screw, to be as 7 to 5.

Having shown the proportions for screws, the next step will consist in considering the different modes of producing the threads alluded to at the commencement of the chapter.

The first process mentioned is that in which dies are used, and one great objection to this method consists in the fact that taps and dies *do not entirely cut the thread*, but form it partly by cutting and partly by squeezing the thread up as it were, under which circumstances it cannot possibly be so strong as a thread left by cutting away the *material cleanly.*

In threads made with dies, it may often be observed that the top of the thread is not sharp and clearly defined, but broken and rough, the metal having been squeezed up on each side, but not in sufficient quantity to fill the die, and make an apparently sound thread. Except for the lightest kind of work, we should never use bolts made with dies, although nuts may be tapped pretty well, as better cutting edges are got on taps than on dies.

In the process of cutting screws with a chaser, the thread is formed by a tool having a number of cutting points, the following part of each point being inclined at the proper angle to produce a thread of the required pitch. The chaser is, in fact, a very thin section of a nut, and to produce the internal screw, the chaser is similarly a very thin section of a screw. In this method the thread is actually formed by cutting away the superfluous metal, and there is no squeezing up, hence screws made with the chaser are far superior to those made in dies.

All bolts for machinery should be cut up in the lathe, and this is a point that engineers should insist upon with strictness.

It frequently happens that instead of a bolt and nut being used, the bolt is screwed into some part of the cast-iron framework, in which case the length of bolt which holds on the thread in the cast-iron should be at least twice the length as calculated by the rules given in the previous part of this chapter, as the resistance to shearing afforded by cast-iron is but half of that afforded by wrought iron. But threads in cast-iron should always be avoided if possible, and it is much better to secure pins that have to be fast in the framing by means of cotters where they can be got in, the cotters passing through a slot hole made through the bolt and the framing.

We must now consider the manipulations connected with screw cutting proper, that is, with the single pointed tool in the lathe, the travel of the tool to produce the required pitch being obtained by means of self-acting feed apparatus on the screw-cutting lathe.

The screw-cutting lathe is furnished with a slide-rest, capable of being traversed along its bed by means of a screw gearing into a nut underneath the rest, this nut being made in two halves so hinged as to admit of being opened, so as to release the screw whenever it is desired to throw it out of gear to wind the slide back to take a fresh cut, &c.

The screw by which the slide rest is worked is called the leading screw, and it should be of the most perfect workmanship, as any irregularities in its construction will be reproduced in all screws cut in the lathe of which it forms a part, being of course magnified in screws of quicker pitch.

The pitch of a screw is the distance measured parallel to the axis of the screw from the centre of the thread to the centre of the next turn of the *same* thread. The word same is italicised in order to prevent mistakes in measuring double or triple threaded screws which are used in quick threaded screws, in order to obtain sufficient strength without excessive depth of thread.

The mode in which this extra strength for a given amount of material is obtained by increasing the number of threads, is simply as follows: Let there be a screw having such work to do that the thread, if single, would require to be three-quarters of an inch square in order to give the requisite bearing and shearing area for one revolution; the bearing area will be equal to the circumference of the screw multiplied by three-quarters of an inch,

and the shearing area will be also equal to the same, the sectional area of the thread will be

$$\begin{array}{rl} \cdot75 & \text{inches side of thread} \\ \cdot75 & \quad ,, \quad \text{width} \quad ,, \\ \hline 375 & \\ 525 & \\ \hline \cdot5625 & \text{square inches.} \end{array}$$

If, however, this thread be replaced by three threads, each one quarter of an inch square, we shall find the same bearing and shearing area in the three small threads as in the one large one, for the total area will be equal to the circumference of the screw multiplied by a quarter of an inch, and by three (which is equivalent to multiplying the circumference at once by three quarters of an inch), and the following calculation will show the sum of the sectional areas of the three threads.

$$\begin{array}{rl} \cdot25 & \text{inches side of thread} \\ \cdot25 & \quad ,, \quad \text{width} \quad ,, \\ \hline 125 & \\ 50 & \\ \hline \cdot0625 & \\ 3 & \text{threads} \\ \hline \cdot1875 & \text{square inches.} \end{array}$$

Thus, by dividing the thread into three, the same strength is obtained with only one-third the sectional area, and, therefore, one-third the weight of metal in the thread. If the screw were divided into four, one-fourth of the sectional area would be required, and so forth.

In the lathe the leading screw is driven by means of

spur wheels, the first being on the back end of the lathe mandril, and the last on the leading screw shaft, the necessary wheels being interposed to obtain the transmission of motion for the different ratios of speed between the mandril and leading screw. The centres of bolts being fixed will of course only admit of being geared direct, while the sum of the radii of the two spur wheels is equal to the distance between those centres, hence it becomes necessary in most cases to interpose a neutral or idle wheel, which transmits motion from one wheel to the other, but does not affect the speed. These wheels which are made in sets and supplied with the lathe are called change wheels, and should be of very accurate construction and machine cut.

We will now find a rule whereby the proportions of the change wheels for the various sizes of screws may be determined for any particular lathe.

Assuming the leading screw to be three-quarters of an inch pitch, each revolution will move the slide rest forward three-quarters of an inch; hence, if it be desired to turn a screw three-quarters pitch, the spur wheels on the mandril and leading screw should be of equal diameter, the diameters of the leading screw and the screw being cut in no way affect the calculation, and therefore need no consideration when determining the sizes of the change wheels.

As one revolution of the screw travels the slide three-quarters of an inch, it follows that with the above arrangement of wheels the screw being cut would be three-quarters pitch, whatever the diameter is, as one revolution of the screw blank in the lathe will in any case be accompanied by three-quarters of an inch travel of the tool. We explain this at large in order to prevent any confusion arising in the minds of those who are not accustomed to the use of the screw-cutting lathe.

If with the three-quarter pitch leading screw it is required to cut a screw of three-eighths of an inch pitch, it is evident the leading screw must make half a revolution to one revolution of the mandril, so as to traverse the slide three-eighths of an inch to each revolution of the screw blank; therefore the diameter of the change wheel on the mandril shaft must be half the diameter of that on the leading screw.

Again, if the pitch of the screw to be turned be one and a half inches, then the leading screw must make two revolutions for each revolution of the mandril, so that at each revolution of the mandril the tool travels one and a half inches; therefore, in this case, the diameter of the wheel on the mandril must be twice that of the wheel on the leading screw.

The following rules will serve to determine the proportions and sizes of the change wheels, the pitch of the leading screw, and that of the required screw being known, and also the diameters of change wheels available:—

RULE.—*To find the ratio of the diameter (or number of teeth) of the mandril wheel to that of the leading screw wheel, divide the pitch in inches of the required screw by the pitch in inches of the leading screw of the latter.*

Let it be required to cut a screw of $1\frac{3}{4}$ inches pitch, the leading screw being $\frac{3}{4}$ inch pitch.

Pitch of required screw.

Pitch of leading screw ·75)1·75(2·3̇3̇
150
250
225
25

RULE.—*The pitches in inches of the leading and required screws being given, also the number of teeth in the change wheel on the leading screw; to find the required number of teeth in the mandril wheel, multiply the number of teeth in the leading screw wheel by the pitch of the required screw, and divide the product by the pitch of the leading screw.*

Let the pitch of the leading screw be $1\frac{1}{4}$ inch, and that of the required screw $\frac{3}{4}$ inch, the number of teeth in the wheel on the leading screw being 40, the required number in the mandril will be thus found.

```
                           40  teeth in leading screw wheel
                          ·75 inches pitch of required screw
                          ---
                          200
                         280
                        -----
Leading screw pitch 1·25)30·00(24 teeth in mandril wheel.
                         250
                         ----
                          500
                          500
                         ----
```

To determine the leading screw wheel the following rule may be used.

RULE.—*The pitches of the leading and required screws in inches being given, also the number of teeth in the mandril wheel, the requisite number of teeth in the leading screw-wheel may be found by multiplying the number of teeth in the mandril wheel by the pitch of the leading screw, and dividing the product by the pitch of the required screw.*

Let the pitch of the leading screw be $1\frac{1}{4}$ inches, and that of the requisite screw $1\frac{5}{8}$ inches, the mandril wheel having 26 teeth.

```
                          26  teeth in mandril wheel
                        1·25  inches, pitch of leading
                         130         screw
                         52
                        26
                       -----
Pitch of reqd. screw 1·625)32·500(20 teeth in leading screw
                        3250          wheel.
                        ----
                           0
```

The easiest practical way to get at the sizes of the change wheels is by taking the ratio, and then seeing what change wheels are at hand to use, thus; let the pitches of the leading and required screws be respectively $1\frac{1}{4}$ inches and $1\frac{3}{4}$ inches; the ratio is thus found.

```
1·25)1·75(1·4
     125
     ---
      500
      500
      ---
```

In this case the ratio is as 1·4 to 1, or as 14 to 10, so the proportion of the numbers of teeth in the mandril and leading screw wheels must be as 14 to 10; for instance, 28 teeth in one wheel, and 20 teeth in the other; and the largest wheel goes on the shaft with the greatest pitch that is on the leading screw, if its pitch is greater than that of the required screw, and *vice versâ*.

Sometimes exceptionally quick pitches have to be cut; as, for instance, we once had to make a screw one foot in diameter, and seven feet pitch. Under such conditions there were no change wheels which would get up the speed enough; hence recourse was had to another expedient:

screw was cut of the quickest pitch of which the lathe was capable, turning out accurate work; this screw was then put in as leading screw, and another quicker screw cut from that, which in its turn was used as leading screw to execute the job required.

In this case the cutting tool had to be turned on its side, the "feed" travel being so much faster than the rate of revolution of the work, that the cut had to be taken longitudinally instead of circumferentially.

Previous to adopting this course, an attempt was made to cut the spiral groove on a planing machine by means of a barrel, having a spiral feather on it as a guide; but that arrangement proved an utter failure, the groove being very much out of truth; but when cut in the lathe, the work gave every satisfaction.

Before taking leave of screw cutting it is necessary to consider the construction of tangent screws, both as to strength and manufacture, as they differ in their conditions materially from ordinary screws working in a whole nut, and the strain of the tangent screw, as a rule, falls upon one side of it only.

It may also be observed that tangent screws are very frequently used for purposes where they are subjected to very heavy strains.

When a tangent screw and worm wheel work in gear together, there is generally one quarter of the circumference of the screw in action, and two turns of the thread, that is to say, we may rely upon two teeth of the worm wheel being fairly in action at one time; but except when the wheel is of great diameter, it is not fair to calculate upon more than two teeth at a time.

It will be found in designing a tangent screw there will *in every case* be a certain diameter, below which it must

not be reduced in order to retain the material necessary for resisting the strain to which it is subject.

The thickness of the thread of the worm will be equal to the pitch multiplied by 0·45, which will allow for clearance, and allowing four tons per sectional square inch as safe shearing strain for a wrought iron screw, the diameter of the screw at the bottom of the thread may be found as follows :—

Rule.—*To find the diameter in inches of a wrought iron tangent screw at the bottom of the thread, multiply the strain in tons which it will be required to sustain by* 0·35, *and divide the product by the pitch of the screw in inches; the quotient will be the required diameter in inches.*

Let the thrust on a tangent screw be 12 tons, the pitch of the screw being 1½ inches, the proper diameter will be found as follows:

```
                 12  tons strain on screw
                ·35  constant
                ---
                 60
                36
               ----
Pitch of screw 1·5)4·20(2·8 nches diameter.
                 30
                ---
                120
                120
                ---
```

This is supposing the work to be exact and the teeth properly set out, but it may happen that all the strain comes upon one thread, in which case the following rule should be used :—

Rule.—*To find the diameter in inches of a wrought iron tangent screw at the bottom of the thread, multiply the strain upon it in tons by* ·7, *and divide* the product by the

pitch of the screw in inches, the quotient will be the required diameter in inches.

Let the strain on a tangent screw be 24 tons, and the pitch of the screw 2 inches.

$$
\begin{array}{rl}
 & 24 \text{ tons strain on screw} \\
 & \underline{\cdot 7} \text{ constant} \\
\text{Pitch of screw } 2) & \underline{16 \cdot 8} \\
 & 8 \cdot 4 \text{ inches least diameter of screw.}
\end{array}
$$

This last is a safe practical rule, and the one we usually rely upon.

There is, however, another matter connected with tangent screws to be considered which has reference to the size of the screw shaft necessary to resist the torsion produced in the transmission of power through it, for of course the least diameter of the screw cannot be less than the diameter of the screw shaft.

To find the moment of torsion on the shaft, we must multiply the thrust on the screw by its pitch, and divide by its circumference, and then multiply by the radius of the screw; but it is generally more conveniently arrived at by calculating the torsion from the force transmitted to the screw shaft at the opposite end to that at which it is given off by the screw.

If the force applied to the screw shaft be known, and also the distance in inches from the centre of the shaft at which it acts, the safe working diameter of the shaft may be found from the following rule:—

RULE.—*To find the diameter in inches of a wrought iron shaft to work under a given strain in lbs., acting at a known distance from the centre in inches, multiply the weight in lbs. by the distance in inches, divide the*

product by 3000, *and extract the cube root of the quotient, which will give the diameter in inches of the shaft.*

Let a shaft be subject to torsional strain equal to 1720 lbs., acting at a distance of ten inches from the centre; as for instance, would be the case if the force were transmitted to a spur wheel on the shaft 20 inches in diameter on the pitch line.

From these data the proper working diameter of the shaft can be determined from the foregoing rule—

1720 lbs. force
10 inches distance from centre

3,000)17,200

5·73

The cube root of 5·73 is 1·79, therefore the shaft should be about two inches in diameter practically.

CHAPTER XII.

MISCELLANEOUS.

UNDER this chapter we purpose making some general observations on certain points which could not conveniently be treated in the previous chapters. In the bulk of the present work we have purposely avoided entering upon general principles, the object being to treat of details, but there are some few mechanical laws that it seems desirable here to refer to in order to simplify modifications of details, which might otherwise appear to alter the principle of a machine in cases where there is but an alteration of form.

One general law of mechanics, applicable to every machine which alters the intensity of force (that is, which, receiving the impression of one force, impresses in its action a force of different intensity), is, that the forces acting at the opposite ends of the machine vary inversely as the velocities with which they act.

We must, in order to illustrate this law, consider the strict meanings of the terms force, work, and power. These terms, or at least one of them (power), are constantly misapplied; the expression a *powerful lever*, has literally no meaning, scientifically speaking, for a lever can only give off the same *power* as is imparted to it, although it *acts with increased force* or *pressure.*

A force or pressure may act *in equilibrio* without producing any motion, in which case it is simply described as force.

Thus, a weight resting on the ground is exerting a certain pressure upon such ground; but the elastic resistance of the ground upwards counterbalances the pressure of the weight downwards, and the result is that the force is acting, but at rest. If the weight or the elastic resistance of the ground preponderated, there would be motion, as one force would overcome the other, hence we find the law of statics:

In order to preserve equilibrium between two forces, the action and reaction must be equal and opposite.

That is to say, the two forces must be equal in intensity and opposite to each other in direction.

If a force act with a certain intensity, and produces motion (such as a locomotive engine drawing a train), then *work* is said to be done, the amount of work being found by multiplying the intensity of the force by the distance through which it passes.

Let a weight weighing 53 lbs. be raised 28½ feet—

$$\begin{array}{rl} 28{\cdot}5 & \text{feet distance through which force acts} \\ 53 & \text{lbs. weight or intensity of force} \\ \hline 855 & \\ 1425 & \\ \hline 1510{\cdot}5 & \text{foot lbs. of work done.} \end{array}$$

The term *power* refers to work done in a given *time;* thus, if one steam engine will lift twice what another will *in the same time*, the former is twice as powerful as the latter. The standard horse-power is equal to 33,000 foot lbs. per minute.

Whether 10 lbs. be raised 50 feet, or 50 lbs. be raised 10 feet, the amount of work done is the same, thus:—

10 lbs. weight	50 lbs. weight
50 feet distance	10 feet distance
500 foot lbs.	500 foot lbs.

Now in this rests the principle upon which the lever and all machines for modifying force depend.

If a common lever be hinged on a fulcrum so that one end is twice as long as the other, a weight of one lb. on the long end will balance a weight of two lbs. on the short end; but it will be found that if the lever be set in motion, the end furthest from the centre will travel twice as far as the short end in a given time, for the length of the arc described by any point in the lever is directly as the distance of such point from the fulcrum. Thus we see the one lb. would pass through two feet while the two lb. passes through one foot, the amount of work done being two foot lbs. at each end of the lever; so that no *power* is gained, for as much is lost in speed as is gained in pressure; and the same obtains in all machinery.

This is a mechanical maxim which it would be well for perpetual motion inventors to consider: that force cannot be increased without loss of velocity; and it may confidently be asserted that no one who *thoroughly understands* the first simple principles of mechanics, would for a moment entertain the notion of a perpetual motion machine.

We shall now give a few general rules for solving ordinary questions that arise in mechanics.

The velocity acquired by a falling body in the first cond of its fall is 32·2 feet, and it falls through 16·1 feet, ing *the average* velocity during the first second, as the

velocity is 0 when the fall commences. Hence we find the average—

$$\begin{array}{rl} 0 & \text{velocity at beginning of fall} \\ 32{\cdot}2 & \quad ,, \quad \text{first second} \\ \hline 2)32{\cdot}2 & \\ \hline 16{\cdot}1 & \text{average velocity or distance passed through.} \end{array}$$

During the second second of the fall the body passes through 32·2 feet by virtue of the velocity already acquired, and through 16·1 feet by the action of gravity during the second second, at the end of which it has fallen through 48·3 feet, and has acquired a velocity of 64·4 feet per second.

The following rules will suffice to calculate the time of fall, &c., of heavy bodies.

Rule.—*To find the height through which a body will fall in feet in a given time, multiply the square of the time in seconds by* 16·1.

Let it be required to ascertain how far a body will fall in thirteen seconds.

$$\begin{array}{rl} 13 & \text{seconds time of fall} \\ 13 & \quad ,, \quad ,, \quad ,, \\ \hline 39 & \\ 13 & \\ \hline 169 & \text{square of time} \\ 16{\cdot}1 & \\ \hline 169 & \\ 1014 & \\ 169 & \\ \hline 2720{\cdot}9 & \text{feet height of fall.} \end{array}$$

Rule.—*To find the velocity in feet per second acquired by a body in a given time, multiply the time in seconds by* 32·2.

Taking the above case where the duration of the fall is thirteen seconds, we find—

32·2	
13	seconds time of fall
966	
322	
418·6	feet per second velocity.

If a body falls through a distance, there is evidently work developed; thus, if a weight of 20 lbs. falls 15 feet, the work developed will be

15	feet fall
20	lbs. weight
300	foot lbs. of work.

If the weight be resisted so that it falls at an uniform rate, then this work will be expended during the falling of the weight; but if on the other hand the weight fall freely, the work will be accumulated within it, and be expended on the first obstruction it meets with. The amount of accumulated work in any body may be calculated from the velocity at which it is moving, by the following rule:

Rule.—*To find the amount of accumulated work in any body in foot lbs., multiply the weight of the body in lbs. by the square of its velocity in feet per second, and divide the product by* 64·4.

Let a train weighing 90 tons be moving at a speed of 40 miles per hour. In the first place, the tons must be *reduced to* lbs. by multiplying by 2240, and the miles per *hour to feet* per second by multiplying by 1·466.

```
  2240 lbs. per ton
    90 tons, weight of train
201600 lbs. weight of train.

1·466 constant
   40 miles per hour
58·640 feet per second velocity.
```

The accumulated work in such train can now be found, first finding the square of the velocity per second.

```
              58·64 feet per second velocity.
              58·64   "     "     "
              23456
             35184
            46912
           29320
           3438·6496 square of velocity
              201600 lbs. weight of train
          20631897600
          34386496
        687729920
64·4)693231759·3600(10764468 foot lbs.
     644
      4923
      4508
       4151
       3864
        2877
        2576
         3015
         2576
          4399
          3864
           5353
           5152
            201
            ...
```

In such a train, moving at 40 miles per hour, there would be accumulated work to the amount of 10,764,468 foot lbs.; hence, in the case of a collision, there would be an enormous amount of work to be expended through a short distance. Assume the train to come into collision with some obstruction which brings it to a stand in 100 feet, then the average pressure against such obstruction would be

$$\frac{100)10764468 \text{ft. lbs.}}{107644 \text{ lbs.}}$$

Which is equal to something more than 48 tons pressure.

From this it is easy to account for the disastrous effects of railway collisions.

Any body set in motion has a tendency to continue moving in a straight line, unless some extraneous force is brought to bear upon it in order to force it out of such straight line. Suppose a body in motion to be kept moving round a centre, it will have a tendency to leave that centre, and keep its original direction, and the force with which it tends to leave the centre, about which it is constrained to move, is termed the centrifugal force of the body.

The centrifugal force of a body may be calculated from the following rule:—

RULE.—*To find the centrifugal force of a body, multiply its weight in lbs. by the square of its velocity in feet per second, divide the product by the radius in feet of the circle in which it moves, and by 32·2.*

Let a body weighing 124 lbs. be moving at a velocity of 11½ feet per second round a centre distant 19 feet from *its centre of gravity.*

11·5 feet per second velocity

11·5 „ „ „

575

115

115

132·25 square of velocity

124 lbs. weight

52900

26450

13225

Radius in feet 19)16399·00(863 first quotient.

152

119

114

59

57

2

The first quotient has now to be divided by 32·2.

32·2)863·0(26·8 lbs. centrifugal force.

644

2190

1932

2580

2576

4

This centrifugal force has a tendency to throw a train of railway carriages off the track when passing round a curve, and to obviate this the outer rail is elevated abo

the level of the inner rail, so as to give the carriages an inclination to slide inwards as it were on an inclined plane, and so counteract the effect, the centrifugal force tending to throw the train outwards.

Rule.—*To find the super-elevation of the outer rail in inches, multiply the gauge in feet by the square of the velocity of the train in miles per hour, and divide the product by the radius of the curve in chains, and by* 397·5.

Let the gauge be 5½ feet, speed 50 miles per hour, and radius of curve 20 chains.

50 miles per hour velocity
50 " " "
Radius of curve 20)2500 square of velocity
397·5)125·00(·314 inches super-elevation.
11925
5750
3975
17750
15900
1850
....

In some cases, where much conicality is given to the tyres of the wheels, that is partially relied upon to assist in overcoming the tendency of the train to be thrown off the metals; but we are of opinion that this should not be done, as, in point of fact, the flange of the wheel should not be in contact with the rail, as its rubbing against the latter must necessarily cause a great deal of friction, which might otherwise be obviated.

CHAPTER XIII.

FIXED JOINTS.

Under the head of fixed joints are included framing joints, and also all others which do not allow of motion between the parts joined. We will, in the first place, deal with framing joints.

In designing the framework of a machine of any description we have to pay particular attention to the vibrations and strain to which it will be subjected; the first joint is that which connects the framing with its foundation. A machine may be so arranged as to stand steadily without any foundation; or its duties may be such as to require anchorage. As an instance of the latter take an ordinary Beam Engine; here in the down stroke, the steam pressing on the top of the piston will with equal force press on the top cover of the steam cylinder, so tending to lift the cylinder bodily from the ground. This lifting action must be resisted by *weight* in the foundations and that of the cylinder itself.

Let the piston be 80 inches in diameter, its area will be 5026·5 square inches; and if the steam works with an initial pressure of 50 lbs. per square inch, the total lifting effort is 251,325 lbs., which would be equivalent to 2,234 cubic feet of brickwork, or 1,470 cubic feet of limestone. Even with the most massive foundations there

must be vibration at every stroke, for as the ground beneath the "cylinder load" is compressed by the weight above it, so it will rise as the weight is partially taken off it, and be compressed again when the lifting action ceases, and thus a tremor is maintained during the action of the engine.

In all cases where it can conveniently be done the framework should be made to take up all the strains, when a more general steadiness may be relied upon. In cases, however, where pulls do come upon the foundations, holding-down bolts secured through anchor plates must be used, the anchor plates being sufficiently large to lift the load without its breaking. The constant alternation of strain and no strain will keep the metallic molecules of these bolts in motion internally, and tend to a flowing of them amongst themselves, and therefore to a permanent lengthening or loosening of the bolts. This may be taken up by tightening the top nuts from time to time; but in order to reduce it to a minimum the working strain on the bolts should be kept well within the limits of elasticity of the material forming the bolts; for good wrought iron the limit is 17,800 lbs. per sectional square inch; but the maximum working stress should not exceed 9,000 lbs. per square inch of sectional area. Taking the above example, and allowing, in addition to the weight of the cylinder and covers, 25 per cent. margin, 2,791 cubic feet of brickwork would be required. The actual pull on the holding-down bolts is 251,825 lbs. (omitting the weight of the cylinder, which is in favour of the bolts), hence the gross area of the bolts must be

9,000) 251,825 lbs.

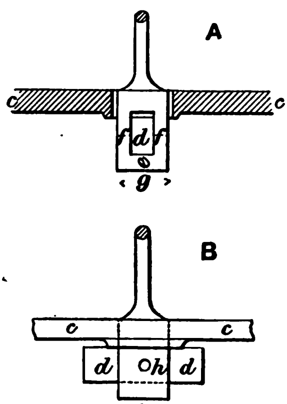
27·925 square inches.

If bolts 2 inches in diameter be used, there will be 8,141 square inches area in each, so that in practice 10 such bolts would be used. Assuming the cylinder load to be 12 feet square, it would require to be nearly 20 feet deep, and through this depth the holding-down bolts must pass.

The anchor plate at the bottom in this case should be made up so as to form one plate, stiffened by ribs on the under side.

The plate and bolts are put down, and the brickwork built up around the latter, sufficient clearance being allowed to enable the bolts to be passed through the holes provided for them in the base plate of the cylinder. We prefer cottering the lower ends of the bolts to screwing them, and using nuts, the cotter end of the bolt being made a large square, as shown in Fig. 23. A and B show a side and an end view as regards the cotter *d*. *c c* is a washer plate, upon which the cotter bears, pressing it upwards against the anchor plate; the length *e* should be at least one and a half diameters of the holding-down bolt; the thicknesses *f f* each three-quarters of the same diameter; and the cotter one diameter thick, and one and a half diameters wide. A steady pin *h* may be used to secure the cotter in its place.

Fig. 23.



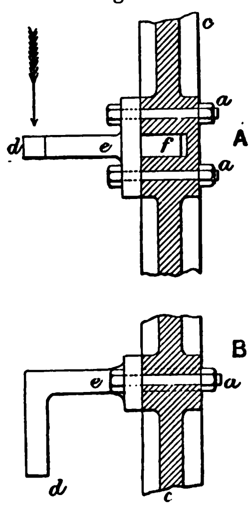
In screwing down the nuts on the holding-down bolts,

they should be sufficiently tightened to give them a good steady hold, and no more, for all the strain thus put upon them is in addition to that legitimately falling on them.

It is a very common blunder among mechanic's labourers to screw bolts and nuts unnecessarily tight, thus unduly straining them, and wringing the screw threads.

We will now pass on to joints in the framing itself, first taking those which connect together the various parts of the frame. Where a joint has a tendency to wriggle, as in the corners of machines containing complicated movements, the bolts are not to be relied upon to resist such wriggling, which, of course, would tend to cut them, and, in the course of time, *would* groove them, and so make the machine shaky. The joints themselves must be made to *fit* together, suitable recesses and protrusions being formed on the different parts, and the only duty left upon the bolts being that of holding those parts in close contact; such a joint is shown in Fig. 24.

Fig. 24.

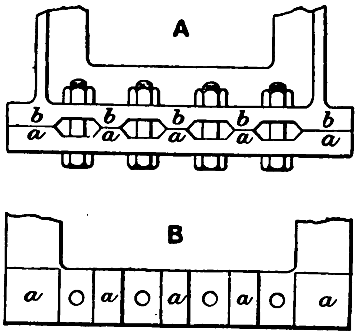


A and B are vertical and horizontal sections; in both the main frame is shown in section at the joint *c*. *e d* is a bracket upon which the strain is supposed to act downwards upon the end *d*, so as to bring a twisting action on the stem *e*, as well as a bending stress; both the twisting *and bending strains* are resisted by the tongue *f*, which

fits accurately in a slot or cavity in the frame casting, up to which the bracket is steadied by the bolts and nuts *a a*.

When one piece of framing meets another in a simple manner, such as an 'A' frame upon a bed plate, the surfaces of contact, if small, are planed all over, but if they be extensive, certain raised parts only are planed, as shown in Fig. 25. A shows an elevation of the joints with planing strips *a a a a a*, on the lower element, upon which rest others, *b b b b b*, upon the upper elements. These strips should be trued in the planing machine, but when that cannot be done they must be chipped true. B is a plan of the lower element with the fillets *a a a a a* on its upper surface.

Fig. 25.

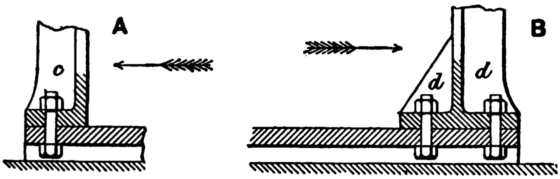


The use of studs frequently occurs in machinery, but where they have to trust to cast-iron threads for their security they are not reliable under heavy strains, as cast iron, being of a pulverulent character, is unsuited for the formation of screw threads, the material of which should be fibrous and tough.

However slight the vibration in a machine, it must be remembered that *there is vibration*, and that it is acting by continual effort to loosen, in some way, the joints, therefore the arrangements must be made with a view to retard such loosening. In Fig. 26 are shown two methods of connecting a side frame with a bed plate, that at A

being inferior to that shown at B. The vibration being lateral in the direction indicated by the arrows, it is evident that the bolt *c* will be liable to a wringing action,

Fig. 26.



a bending backwards and forwards, whereas, with the two bolts *d d*, there will be only exerted a pulling strain upon each bolt alternately.

We have given rules for proportioning keys in shafts at page 39, *et seq.*, but some more general remarks are here called for in relation to the connection of revolving and oscillating elements with their shafts.

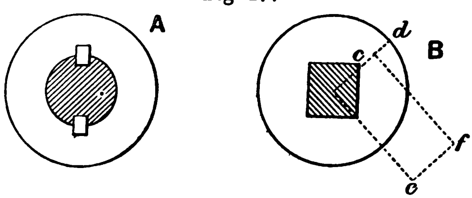
It is evident that oscillating arms, &c., will tend more to dislocate their joints than those that continue to revolve in one direction, for in the latter case the strains established at starting are, in character, if not exactly in intensity, maintained unchanged until the machine is stopped, but an oscillating member is constantly changing the nature of the strain upon its connections and so tending to wriggle them loose.

The double key shown at A Fig. 27, will certainly have less tendency to burst the boss than will the square on the arbor shown at B. The former is best for the boss, the latter for the shaft.

Proceeding with the case of the square, rupture of the boss will occur along a line *c d*, running from one corner of *the square*, radially or nearly so, to the periphery of

the boss. Now if the material were perfectly uniform in texture, or homogeneous as it is commonly called, and the workmanship perfectly true, the boss should simultaneously break at all four corners; but practically this will not obtain; the weakest corner will first give way, and so the boss will be destroyed in detail, therefore each corner should be made of a strength determined upon this supposition. The breaking force acts to tear apart the section *c d* of metal, the force upon that section being equal to the force acting at the end of the arm or radius of the wheel to which the boss belongs, multiplied by that

Fig. 27.



radius divided by the line *e f*, which is the distance from the centre of the arm or wheel to the centre of the section *c d*. For instance, let the arm or radius be 3 feet, the force acting at the end of it and at right angles to its length two tons, the distance *e f* 3 inches (the shaft being 4 inches in diameter and the boss 2 inches thick). Three feet are 36 inches, hence we have—

$$\begin{array}{r} 36 \text{ inches} \\ 2 \text{ tons} \\ \hline 3)72 \\ \hline 24 \text{ tons tension on section } c\ d. \end{array}$$

If, then, we take 3 tons per square inch as a safe working strain, we shall require 8 square inches, so that the width of the boss should be 4 inches. These figures are given for wrought iron. For cast iron the tensile strain should not exceed one ton.

CHAPTER XIV.

REVOLVING AND OSCILLATING JOINTS.

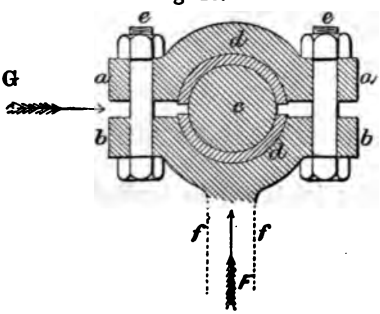
Revolving and oscillating joints may be typified by two well-known examples—the former by the connection of the connecting-rod of an ordinary horizontal steam engine with the crank pin through which it drives the crank, and thereby the main shaft; the latter by the connection in the same class of engine of the piston rod head with the connecting-rod. For it will be observed that the crank pin revolves within the bearings of one end of the connecting-rod referred to, and the other end of the same connecting-rod oscillates upon a dead centre firmly attached to the head of the piston rod.

We will take another example, that of the oscillating engine so commonly used in small steam vessels. In this class of engine there is no connecting-rod, the head of the piston rod working immediately upon the crank pin, while the oscillating joint is illustrated by the movement of the trunnions of the oscillating cylinder within their bearings.

In designing the bearings within which the pins and shafts revolve or oscillate, it is necessary to have regard to the directions of the strain upon such bearings. Referring to Fig. 28, *c* represents in section a crank pin working in brass bearings *d d*, *a a* being the cap and *b b*

the head of the piston rod; it will be observed that between the bearings *d d* and the elements *a a* and *b b* a space is allowed in order that wear may be compensated by tightening up the bolts *e e*. Assuming the strains to act in the direction indicated by the arrow F, it is evident that this arrangement will answer the designer's purpose; but if the strains are in the direction indicated by the arrow G, the compensating arrangement would be inefficient, as the wear upon the bearings *d d* would be horizontal, and therefore could not be taken up by the vertical action of the bolts *e e*. Assuming the dotted line *f f* to represent a piston rod or any similar element, the construction here shown is correct.

Fig. 28.



We will now proceed to show how to determine the dimensions of the various parts of this joint, regarding it as a revolving joint, *c* being a crank pin, and the lines *f f* as representing the piston rod.

RULE.—*To find the diameter of the crank pin, assuming it to be of wrought iron, divide the maximum force in pounds, acting in the direction* F, *by* 5280, *and take the square root of the quotient; the result will give the diameter of the crank pin in inches.*

This does not apply to any especial case, but is a general rule for such joints, whether the pin *c* is carried on a crank making a complete revolution or on one working a

rocking shaft. The crank pin is assumed to be carried on *one* crank. If it drives a pair of cranks, the factor 5280 must be replaced by 10560.

To illustrate this rule, we will assume a case in which the force acting in the direction F is 14,000 lbs.; then—

Force.

```
5280)14000(2·651
     10560
     -----
      34400
      31680
      -----
       27200
       26400
       -----
         8000
         5280
         ----
         2720
         ....
```

We now extract the square root of 2·651 :—

```
         .  .  .
      1)2·6510(1·628 inches diameter.
        1
       ----
 2·6)  165
       156
      -----
 3·22)   910
         644
        -----
 3·248) 26600
        25984
        -----
          616
          ...
```

Thus we find the diameter for the pin carried by a single crank to be 1·628 inches, the force to which it is subject being 14,000 lbs.

This one illustration of the rule will be sufficient to illustrate its working.

We will now determine the necessary diameter of the bolts. As the shearing strength of wrought iron is practically equal to its tensile strength, and the strain upon the crank pin *c* is one tending to shear it through, while the strain upon the bolts is tensile, the sum of the areas of the bolts must be equal to the area of the crank pin if on a single crank, or to twice its area if the crank pin be carried by double cranks; hence we have the following rule.

Rule.—*To find the diameter of each bolt divide the diameter of the crank pin on a single crank by the square root of the number of bolts.*

Following the same example we have, assuming 2 bolts, to divide 1·628 inches by the square root of 2, which is 1·414, as follows:—

```
1·414)1·628(1·151 inches.
      1·414
      -----
        2140
        1414
        ----
         7260
         7070
         ----
          1900
          1414
          ----
           486
           ...
```

which will be the requisite diameter for each bolt.

Of course it must be borne in mind that these decimal numbers cannot in most cases be exactly worked to, but in no case should a bolt less in diameter than that calculated be used. Some, in their inexperience, might fall into the error of adopting bolts of the size *nearest* to the calculated dimensions, whereas they should adopt *the nearest size above* the calculated dimensions; for instance, in the present case we should adopt bolts $1\frac{3}{16}$ inch diameter.

Where the crank pin is carried on a double crank, twice the number of bolts given by the above rule must be used.

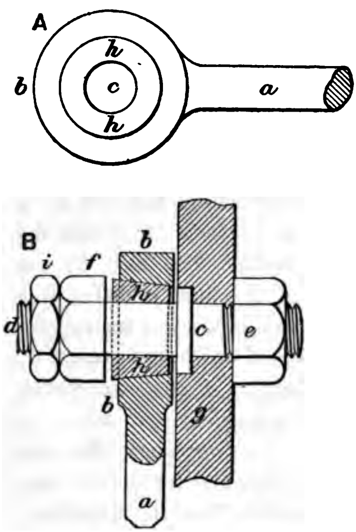
In these moving joints the very greatest care should be used to insure the correct proportioning of the bolts as to their heads and nuts, and also as to their fitting into the holes *bored* in the elements *a a* and *b b*, Fig. 28. We use the term bored advisedly, as if the hole is drilled the drill used should be such as will produce a truly cylindrical hole, and not one of those (of which, unfortunately, there are too many) which straggle through the metal, much as an earthworm may do through the ground.

In proportioning the bolts for moving joints, the nuts should in height be at least one diameter for the *best* material, but it is preferable to make them $1\frac{1}{2}$ diameters, unless a "lock nut" be used, that is a thinner nut on the same bolt, the height of the two being one and a half diameters of the bolt. The advantage of using two nuts is that one may serve to *slightly* jam the other upon the thread, and so reduce the tendency to working loose. The heads of the bolts should in height be equal to one diameter of the bolt.

For the thickness of the bearings *d d* we can give no definite rule, that which is requisite depending entirely upon the quality of material used and the lubricants available.

In small machinery particular attention should be given to such revolving or oscillating joints as those which consist of an eye working upon a pin or "dead centre," as shown at Fig. 29. At A is shown the end of a bar *a*, rocking upon a dead centre *c*, which is accurately turned to fit the eye *b*; the dead centre *c* is shown with the nuts removed.

Fig. 29.



At B is shown a vertical section of a similar joint, the letters corresponding to those at A, except the additional parts illustrated. It will be seen in this section (B) that the dead centre is in the form of a stud, being secured to the frame-work *g* by the nut *e*, which tightens up the collar *c* against the other side of the frame.

In this arrangement a conical collar *h h*, is fitted upon the dead centre, and upon this collar the rocking arm works, and such wear as occurs can be compensated by tightening the nut *f* on the end *d* of the dead centre, and securing it by the lock-nut *i*. In this case the use of the lock-nut is to allow the connection to be steady, without jamming up the working parts—an important matter in respect to wear and depreciation of machinery.

CHAPTER XV.

GUIDES FOR SLIDING ELEMENTS OF MACHINERY.

THE vibration caused by the action of all descriptions of machinery is, in most instances, due to insufficient strength in those guides to which this chapter applies.

It is almost impossible to give set rules for the sizes of such guides, as the tendency to vibration varies so much under causes that can scarcely be predicated.

We must in such cases be therefore guided ourselves by the practical experience we have acquired, remembering the advice of Smeaton, that we can only use every precaution against accident when we cannot accurately determine the force with which we have to deal, for although his words applied to the construction of light-houses, they are equally applicable to our present subject.

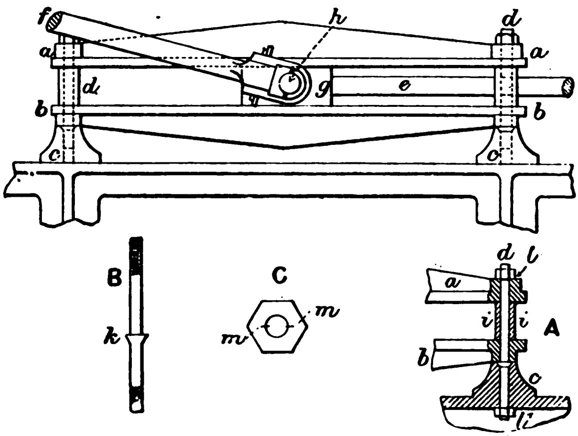
We have not only to consider the strength of the guides themselves but also the stability of the frame-work to which they are attached. In lightly built iron steam launches the framework is often unequal to the strains thrown upon it by the engines in their re-actions upon the parts supporting them.

In Fig. 80 is shown a pair of guides for a small steam engine; *a a* is the top and *b b* the bottom guide bar. These are secured to bosses *c c* upon the main framework

K

of the engine by means of bolts *d d*, of which one is shown more in detail at A. In this view the ends of the guide bars and of the main frame or bed-plate are shown in section, in order more clearly to illustrate the mode of connection by the bolts *d*. These bolts without their nuts are of the form shown at B, having a conical collar

Fig. 30.



at *k* and being screwed at both ends to receive the nuts *l l*, shown on view A. The boss *c* is bored out so as to be a close fit for the bolt or stud *d*, and at the top it is countersunk to receive the conical collar *k*; the shorter end of the stud *d* being passed through the hole in the boss *c*, it is secured in its place by the nut *l*, and if there is any danger of slipping, a lock nut may be placed outside it, or it may be secured from turning by making a couple of light notches *m m* (at C) with a bluntish cold metal chisel. This will sufficiently burr up the two threads, that on the bolt and that in the nut, to prevent

the nut from working loose. It is not advisable to cold rivet the end of the stud over the nut, as in that case there would be no getting the stud out at any future time when it may be desired to replace the guide bars.

The stud having been thus firmly secured to the bottom frame, and the guide bars having had their ends accurately bored to fit the upper part of the stud *d*, the end of the lower guide bar *b* is passed on to the stud until it is stopped by the boss *c*; there is then passed over the stud a distance ring, shown in section at *i i*. This is to maintain the proper distance between the top and bottom guide bars, and the ring will therefore be in length a *shade* more than the thickness of the guide block *g*, into which is cottered the end of the piston-rod *e*. The guide block *g* has dead centres upon each side, on which the connecting-rod *f* oscillates, that rod being formed with a forked end wide enough to clear the guide block and long enough to clear the ends of the guide bars when the piston is at the bottom of the cylinder.

We must now give a rule for finding the strength of the guide bars.

The strain on the guide bars will be greatest when the guide block is in the centre of their length, the maximum strain being on the bottom guide when the crank is above the crank shaft, and on the upper guide when the crank is below the crank shaft.

RULE. *To find the strain on the guide bar at right angles to its length, multiply the* MAXIMUM *gross pressure on the piston by the length of the crank (centre to centre of main shaft and crank pin), and divide the product by the distance of the centre of the main shaft from the centre of the pin h* (Fig. 80) *when the crank has made a quarter of a revolution.*

SUPPLEMENTARY RULE.—*To find the required distance*

from the crank shaft to pin h, square the length, between centres, of the connecting-rod, and from the product subtract the square of the length, between centres, of the crank; the square root of the remainder will be the length required.

Let the length of the connecting-rod be 34 inches and the length of crank 9 inches.

```
    34 inches length of connecting-rod
    34   "     "    "       "       "
   ----
   136
  102
  ----
  1156 square of length of connecting-rod
    81 square of (9 inches) length of crank
  ----
3)1075(32·78* inches length required.
   9
  ----
62)175
   124
  ----
647)5100
    4529
   -----
6548)57100
     52384
     -----
      4716
      ····
```

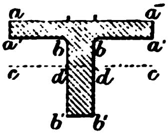
Having thus ascertained the required distance from the crank shaft to the guide block centre, we can show the method of applying the rule. Let the engine have a *cylinder 12* inches in diameter, and the *maximum* pressure

of steam be taken at 80 lbs. per square inch. The area of the cylinder multiplied by the pressure will give the maximum force upon the piston rod, from which we have to find the pressure on the guide blocks.

```
     12   inches diameter of cylinder
     12     ,,       ,,       ,,
    ----
    144   square of do.
   ·785   multiplier
   ----
    720
   1152
  1008
  ------
  113·04   square inches area of cylinder
       80 lbs. per square inch
    ------
   9043·2 lbs. maximum gross pressure
        9 inches length of crank pin
   ------
*
32·78) 81388·8(2482·8 lbs. pressure on the guide bar
       6556
      ------
      15828
      13112
      ------
      27168
      26224
      ------
        9440
        6556
       ------
       28840
       26224
       ------
        2616
        ....
```

We must now give a rule by which to find the strength of a guide bar. The rule given beneath assumes that the castings are made of sound grey cast iron, of uniform texture, and of section shown in Fig. 31. In the first place it will be necessary to determine the position of the line c c, passing through the "centre of gravity" of the section.

Fig. 31.



This section is to be regarded as consisting of the two parallelograms $a\ a\ a^1\ a^1$ and $b\ b\ b^1\ b^1$. To find the position of the line c c we have the following:—

Rule.—*Multiply the length a a, by half the square of the depth $a\ a^1$; multiply the depth $b\ b^1$, by the thickness $b^1\ b^1$, and by $a\ a^1$ plus half $b\ b^1$; add the products together for a total; multiply a a by $a\ a^1$ and $b\ b^1$* by *$b^1\ b^1$; add these products together, and by their sum divide the total previously obtained; the quotient will be the distance of the line c c from the edge a a of the section. All dimensions should be taken in inches.*

Let the length a a be 4 inches, the thickness $a\ a^1$ $\frac{7}{8}$ inch (·875 inch), let $b\ b^1$ be 3 inches, with a thickness $b^1\ b^1$ of $\frac{3}{4}$ inch (·75 inch), then for the first total we have—

$$
\begin{array}{rl}
2)\cdot7656 & \text{square of } \cdot875 \text{ inches} \\
\hline
\cdot3828 & \\
4 & \text{inches length of } a\ a \\
\hline
1\cdot3312 & \\
\hline\hline
\end{array}
$$

·75 inches thickness $b^1\,b^1$
3 inches depth $b\,b^1$

$$
\begin{array}{r l}
2{\cdot}25\dagger & \\
2{\cdot}375 \quad \cdot\;\cdot\;\cdot\;\cdot & \left\{\begin{array}{l} 0{\cdot}875 \text{ inches } a\,a^1 \\ 1{\cdot}500 \text{ inches half } b\,b^1 \\ \hline 2{\cdot}375 \end{array}\right. \\
\hline
1125 & \\
1575 & \\
675 & \\
450 & \\
\hline
5{\cdot}34375 & \\
1{\cdot}3312 & \\
\hline
6{\cdot}67495 & \text{first total.}
\end{array}
$$

Now, for the second total, we have $a\,a$ multiplied by $a\,a^1$ equals 3·5, and $b\,b^1$ multiplied by $b^1\,b^1$ equals †2·25 square inches, making a total of 5·75 square inches :—

$$
\begin{array}{l l}
5{\cdot}75)6{\cdot}67495(1{\cdot}160 & \text{inches distance of line } c\ c \\
\phantom{5{\cdot}75)}575 & \text{from edge } a\ a, \text{ and cutting} \\
\phantom{5{\cdot}75)}\overline{924} & b\ b^1 \text{ in points } d\ d. \\
\phantom{5{\cdot}75)}575 & \\
\phantom{5{\cdot}75)}\overline{3499} & \\
\phantom{5{\cdot}75)}3450 & \\
\phantom{5{\cdot}75)}\overline{495} & \\
\phantom{5{\cdot}75)}\cdots &
\end{array}
$$

The total depth of the rib is 3·875 inches, hence the line $c\,c$ will be 2·715 inches from the edge $b^1\,b^1$, and from this lower portion the strength is to be calculated. The safe working pressure on the guide bar will be found from the following :—

RULE.—*Multiply* 3200 *lbs. by the thickness* b^1 b^1, *and by the square of twice the depth* d b^1, *and divide by the length of the guide from centre to centre of holding bolts, all in inches; the quotient will be the safe working pressure at the centre of the guide.*

Taking the above dimensions we have, if the length of guide is 24 inches—

2·715 inches depth d b^1
2
―――
5·43
5·43
―――
1629
2172
2715
―――
*29·4849 square of twice d b^1.

3200 constant
·75 inches thickness b b^1
―――
16000
22400
―――
2400
*29·48
―――
19200
9600
21600
4800
―――
Length of guide in inches. 24 { 8)70752·00
―――
3)8844·00
―――
2948 lbs. safe pressure on guide bar.

A simple rule for unsymmetrical beams of this description cannot be given, but of course by inverting that just exemplified, the depth $d\,b^1$ being given, $b^1\,b^1$, the thickness, may be found; then the upper part must be arranged so as to keep the centre of gravity on the line $c\,c$.

The holding bolts will invariably have a great excess of strength; thus we should not think of putting in, in this case, bolts less than $\frac{7}{8}$ inch diameter, which would give $\frac{3}{4}$ inch diameter at the bottom of the thread, giving an area of 0·44 square inches per bolt, or for the two bolts—one at each end—0·88 square inch; allowing, then, for safe tensile strain under vibration, 7,000 lbs. per sectional square inch, the working strain on the bolts may be as high as—

$$\begin{array}{r} 0{\cdot}88 \text{ square inches area} \\ 7000 \text{ lbs. per sq. inch} \\ \hline 6160{\cdot}00 \text{ lbs.} \end{array}$$

The common practice for the size of nuts is to make them of a height equal to the diameter of the corresponding bolt, in order to obtain equal strengths; but we should here point out that when studs tapped into cast iron are used, the length tapped should be at least equal to two diameters of the bolt, in order to give equal strength, and even then threads in cast iron are not so reliable as they might be.

Very great care should always be used to obtain good castings for machinery of all descriptions, as a single flaw may cause the destruction of the whole machine, accompanied perhaps by loss of life.

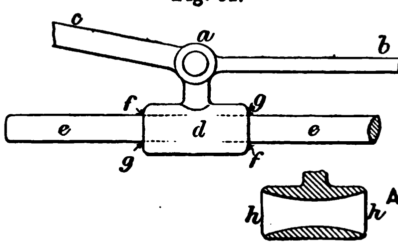
We must say that we feel the more anxious on the subject of cast iron from noticing the quantity of cinder

trash that has been palmed off in bridge columns and other work, where competent and reliable inspectors are not employed.

The faces of the guides must of course be properly planed and faced-up and the guide block accurately fitted to them, so as to move with undue friction and yet without any jumping. The faces of the guides must be exactly parallel and carefully adjusted to the guide block.

In Fig. 32 we illustrate another form of engine guide, which, although it looks *natty*, we unhesitatingly condemn, and for several reasons, as will appear further on:—

Fig. 32.



a b is the end of the piston rod fastened to a casting *d*, and being embraced at the end *a* by the forked end of a connecting-rod *a c*; *e e* is the guide bar upon which the casting *d* slides to and fro.

Now it is evident that *no lateral* steadiness is afforded to the piston rod in this arrangement, as the casting *d* is at liberty to revolve upon the guide bar *e e*; hence, if the piston and connecting-rods are not in strict alignment they will spring at the joint *a* at every stroke. Then again, there is the question of wear. It is obvious at sight that the stress upon the casting *d* produces an effect *other than that* of merely sliding it upon the guide bar *e e*—

there is a tendency to rock the casting; during the out-stroke of the piston rod the pressure will be on the parts of the internal edges at the parts *f f*, and on the return stroke the pressure will be on the parts *g g*.

The result of such constantly changing pressures must be to wear the hole through the casting oval at the ends, thus making it in vertical section trumpet-shaped at end, as shown at *h h* in view A. When this occurs the arrangement must become very rickety, especially in high speed engines; nor is there any method of compensating for this wear.

Sometimes the piston rod is extended sufficiently beyond its connection with the connecting-rod to allow of the end working through a ring in order to secure its rectilineal action ; this arrangement is preferable to the above, as the piston rod is steadied in every direction, nor is there any rocking effort on the guide; but here again the tendency is to wear the hole in the guide to an oval form, and there is no adjustment to compensate. When this arrangement is adopted the connecting-rod must be forked in order to allow of its clearing the guides.

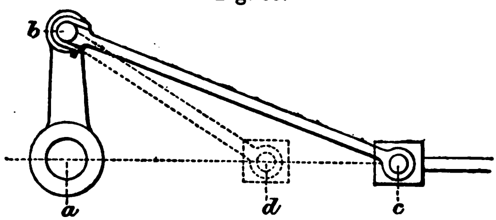
Here, however, we have no guidance tending to prevent twisting, so that if from any unequal wear the piston rod has a tendency to twist, some wrenching of the forked connecting-rod may ensue.

This seems a fitting place to show why the oscillating member of a pair, such as piston rod and connecting-rod, should be made as long as possible.

In Fig. 33 let *a* be a shaft carrying a crank *a b*, which is actuated through a connecting-rod *b c*, by a rectilinearly moving rod of which the guide block end is shown at *c*. A reference to our rule for pressure on the guides will show that if the crank length remain fixed or constant

while different lengths of connecting-rod are tried, the pressure on the guide block will vary. It varies inversely as the distance between *a* and *c*; if, therefore, the connecting-rod *b c* be replaced by a shorter one *b d*, the distance referred to is shortened from *a c* to *a d*, and by so much inversely is the pressure on the guide bar increased; thus if *a d* be one half of *a c*, the use of the connecting-

Fig. 33.

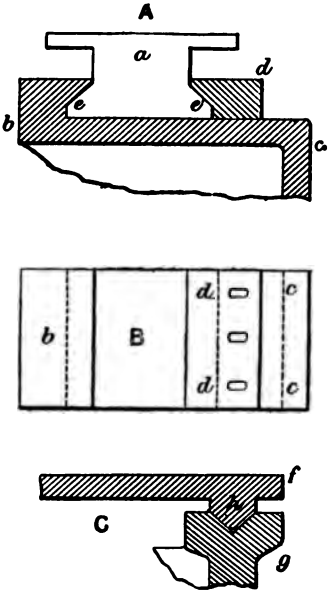


rod *b d* will be attended by twice the pressure on the guide bar (and therefore twice the friction) that will occur when the connecting-rod *b c* is used. The additional friction will also cause loss of efficiency, for a correspondingly less quantity of power will be transmitted through the connecting-rod to the crank.

There is a very important class of guides for the sliding tables of certain machine tools, to which we will now give our attention. A generic form is shown in Fig. 34. A is a section, and B a part plan of this guide. It will be seen that the sliding element *a* (which may be the slide-rest of a lathe) is formed with wedge-shaped flanges at the bottom, on each side, shown at $e\,e^1$. The bed upon which this element moves is shown in transverse section at *b c*, A, and in plan *b c*, B. The bed is made with *one* under-cut, into which *e* is fitted so as to slide without any jar or

shake; and it is held against this guide by the strip *d*, which is adjustable, being secured to the bed by bolts passing through the slotted holes, which will allow of the strip enclosing the flange e^1 being brought as close as may be desired to the element *a*, and from time to time adjustments may be made by shifting the strip *d* to compensate for wear. There is another description of guide commonly used in planing machines, which is shown in section at C; *f* shows a portion of the bed of the machine with a *V*-piece, *h*, cast on to its side. This *V*-piece fits and works in a similarly shaped groove in the bed *g* of the machine. This arrangement will be self-adjusting unless unequal wear should occur to a great extent.

Fig. 34.



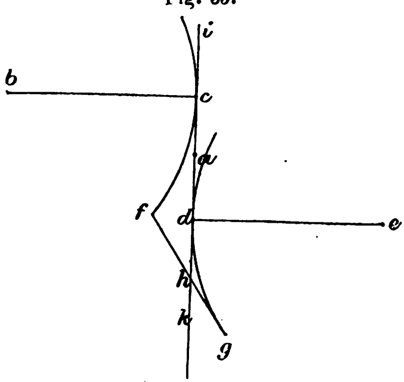
The foregoing examples may be regarded as typical of all rectilineal *sliding* guides; and when properly constructed and proportioned, they are absolute parallel motions, but they occasionally entail considerable friction.

CHAPTER XVI.

PARALLEL MOTIONS.

ALTHOUGH the contrivances described in the previous chapter secure parallelism of motion, yet they are not what are commonly understood as "parallel motions," the latter being formed usually of articulated rods. One of the simplest forms of this class of motion is shown at Fig. 35. *b c d e* represents the system of bars called the parallel motion; *b* and *e* are "dead centres" attached to some part of the framework of the engine or machine in which the motion is to be employed; the free ends *c d* of the links *b c*, *e d* are connected by the link *c d*, and in this link there will be a point *a* which will move in a straight

Fig. 35.



line. Thus if to such a point the head of a piston or other rod be connected its motion will be rectilineal.

In the example shown the links *b c* and *e d* are of equal length, in which case the point of rectilineal movement will occur at the *centre* of the link *c*. If the links *b c*, *e d* are of unequal lengths, the point of rectilineal movement will not be in the centre of the connecting links.

But to return to the Fig. We will explain *why* the point *a* moves in a right line, so that being understood, the principle upon which to design articulated parallel motions generally may be clearly comprehended.

If we move the link *c d* so that the point *a* descends to *h*, we shall cause the links *b c* and *e d* to move to the positions *b f* and *e g*, *c f* being equal to *d g*. At the same time the ends *c* and *d* of the links *e d* will deviate one on each side of the right line *i k*, and to practically the same extent, so that the centre or mean point will move in a straight line. Although this is not in the abstract absolutely exact, yet for such small arcs as occur in those motions practically applied the movement may be regarded as sensibly parallel.

To enter at length into the *theory* of parallel motions generally would be out of place in the present work, which is devoted to dealing with details in a purely *practical* sense.

It will be evident on the slightest consideration that the element of strength need hardly be considered in connection with articulated parallel motions of the class here described; the strains upon them will be practically nothing compared to the strength which the links must necessarily possess if they are made strong enough to carry their own weight without deflecting; hence it becomes a mere question of designing the various bars so as to

bear some reasonable proportion to the rest of the machine.

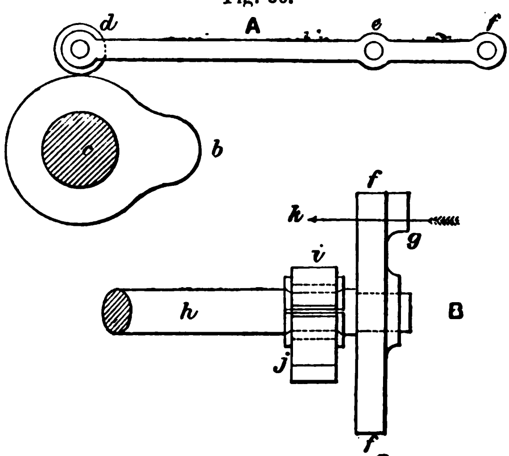
It would be tedious, and in fact almost useless, to attempt to give rules for determining the lengths of the various elements of parallel motions, as they can always be best set out graphically.

CHAPTER XVII.

CAMS AND THEIR ADJUNCTS.

In designing cams and the elements necessarily associated with them, we encounter a very irregular class of strains, due to the intermittent action of these contrivances.

Fig. 36.



Let us consider the strains on a cam of the form shown at A, Fig. 36. This is a common form of "edge-cam,"

having upon its periphery one camber, *b*. *e* is a dead centre, upon which is mounted a lever *d e f*, having at *d* a roller which rests in contact with the edge of the cam, *f* being the end of the lever from which the work is done at every impulse given to the end *d* by the camber *b*. Now, it is evident that there is no strain on the shaft *c*, beyond that due to the weight of the cam and shaft during each revolution, except that part which is performed during the action of the cam upon the lever. Therefore, it is evident that excessive wear will accrue to that part of the shaft which is diametrically opposite to the camber *b*, and the greatest wear of the bearings will be on a portion of the periphery opposite the roller *d*; the arc both on the shaft and in the bearings will depend in magnitude upon the distance through which the camber acts. Thus, for instance, if the cam be in the form of a simple excentric, such as that adopted in the old-fashioned "crocodile" shearing machines, there is a varying strain throughout the revolution, for there is always the weight of the ponderous lever on the cam, and this will give in some positions a twisting strain as well upon the shaft; when the "cut" comes on the strain is nearly radial. This form so well exemplifies varying strains of different kinds occurring in the same element that it well deserves being described as a representative case.

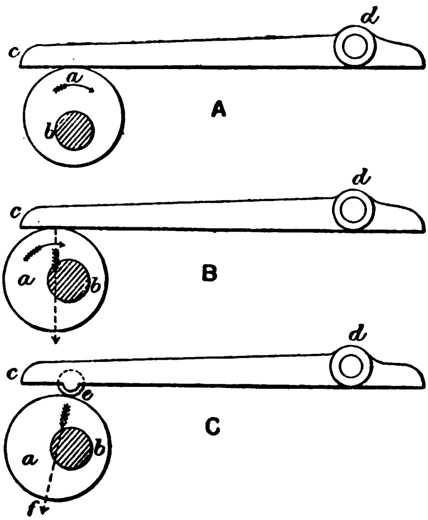
Figure 87 represents outline elevations of a cam and lever of the class referred to in different positions—*a* is the cam, or excentric, mounted upon a shaft *b*; *c d* is the one end of the lever of which the other carries the shearing edge.

In the view A, the cam is at the top, and therefore the shearing edge being depressed to its lowest position the "cut" is complete; here it will be observed that the

strain upon the shaft *b* is simply radial, and, therefore, if it is held in between close bearings (as in all cases it should be) the strain will be a shearing strain. Rules for determining the areas necessary to sustain the various strains will be given farther on.

Proceeding now to consider the position shown in the view B, when the cut is commencing the strain acts in

Fig. 37.



the direction of the arrow, that is, through the geometrical centre of the excentric cam, thereby putting a twisting strain upon the shaft *b*; and, on the opposite side, when the cam has made half a revolution the twisting strain will occur, but in an opposite direction; so that the shaft *b* is twisted alternately in opposite directions, being sub-

jected to two such twists in each revolution. The cam revolves in the direction indicated by the curved arrows.

The pressure on the end of the lever being known, the necessary working diameter of shaft may be determined by the following:—

Rule.—*Divide the radial strain in lbs. upon the cam shaft by* 7,000*; the square root of the quotient will be the required diameter in inches of the shaft.*

This will be the diameter at the smallest part.

Let there be on the cam a pressure of 16,000 lbs.

7,000)16,000 lbs. pressure on cam

2·285

Of which the square root is 1·51 inches.

We must, however, also see that we have sufficient area to resist the twisting strain, but we shall almost invariably find that the diameter calculated by the above rule will also be sufficient to resist the twisting strain.

The view C shows the case where the lever is fitted with a friction roller *e*; here the arrow *f* indicates the direction in which the pressure comes upon the cam.

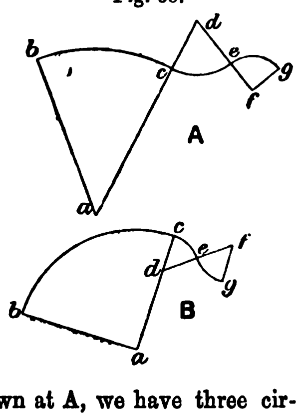
Returning to Fig. 86, as the greatest pressure on the bearings is opposite the roller *d*, care should be taken that the plummer block is in that position, so as to afford a solid resistance; for if this pressure comes upon the cap the whole strain will be taken upon the bolts, and, although of course these should be made of sufficient strength, yet it is best always to provide the most solid bearing obtainable.

At B is shown another form of cam, known as a face cam; *h* is the shaft carried in bearings held in a plummer block *j* covered with a bearing in a cap *i*. *f f* is a disc carrying a camber *g*, which actuates a lever at every revo-

lution; here the force acts in the direction of the arrow *k*, and, therefore, tends to bend the shaft *h*; the shaft being amply strong to resist such bending, the cam must be made with a long bearing on it, say a boss the hole bored through which is in length equal to one and a half diameters of the shaft, so that it may not wear loose.

The variety of mechanical movements obtainable by the use of cams is infinite, but these elements require great care in setting out, and on account of their irregularity of pressure on cams of all descriptions it is advisable that they should be made of hard material, or of wrought iron, case hardened; care should also be taken that the curves forming the periphery of the cam, or in the case of face cams, the curve of the cambered parts, be so adjusted to each other as to obviate any jerking movements. It may be as well here to show how two curves may be joined so that the passage from one to another may be perfectly smooth. We will assume that the curves are circular arcs.

Fig. 38.



In Fig. 38 are shown portions of the contours of two kinds of cams, that at A being part of an edge cam (similar to that shown at A, Fig. 36), and that at B part of a heart-shaped cam.

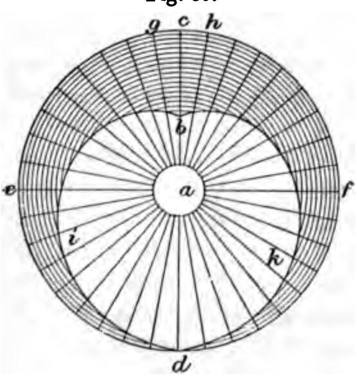
Taking first the form shown at A, we have three circular arcs, *b c*, *c e*, *e g*; the centre from which the arc *b c* is struck is *a*; from *a* is drawn the radius *a c*, which

is produced as far as may be required. On this produced line *must be* the centre from which the centre arc *c e* is struck, in order that the junction of the curves at *c* may be accurate; the centre of *c e* is shown at *d*; from *d* is drawn the radius *d e*, which also is produced as far as may be necessary for the centre *f* of the arc *e g* to be marked.

Passing now to the view B, we have the three arcs *b c*, *c e*, *e g*; the first arc *b c* is struck from the centre *a*, and on the radius *a c* is located the centre *d*, from which is struck the arc *c e*; the radius *d e* is produced far enough to reach the centre *f*, or the arc *e g*.

From this it will be observed that in all cases where two circular arcs have to be joined, in order to secure a smooth path from one to the other, it is necessary that the centres from which the two segments are struck should be located in a straight line passing through the junction of the curves.

Fig. 39.



In the construction of small cams it is perhaps most convenient to set out the contour on the work itself, but whether it be set out so, or on a template, a systematic course must be pursued. In Fig. 39 is shown a method of setting out one class of cam, that in which the lever or rod upon which the cam is arranged to act is required to be *uniformly* raised during one-half of a revolution and

allowed to fall uniformly during the following half revolution, that is to say, the travel of the lever away from, or towards, the centre of revolution of the cam must be constant for any given portion of a revolution.

In the figure *a* is the shaft upon which the cam is to be fixed, *b* shows the part of the periphery of the cam nearest to its centre of revolution, and *d* the point farthest therefrom.

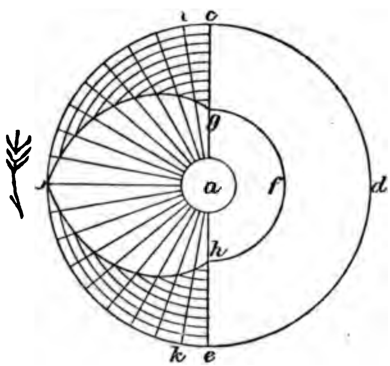
From the centre *a* we describe with the radius *a d* the circle *d e c f*, and this circle we divide into thirty-six parts, drawing from the points of division radial lines as shown; we, therefore, have eighteen divisions for each semicircle or half-revolution, during which the lever-end or roller resting at *b* is to be lifted through the distance *b c*. Divide *b c* into eighteen equal parts, and from the centre *a* describe arcs of circles to radii determined by such division as shown. Thus, the first arc is bounded by the radii *g a* and *h a*, the next arc will reach to the next radius on each side, and so on until the outer circle *d e c f* is reached, then the points at which the several arcs meet the radii will be points in the periphery of the required cam, the contour of which will consist of two spirals *b i d* and *b k d*.

If a large number of cams are required of one size and form it will be most economical to prepare carefully a special geometric chuck, so that the bulk of the cams may be turned in the lathe.

We have with this cam an uniform movement, but it is advisable also to give an example of one acting intermittently, as shown at Fig. 40. *a* is the shaft upon which the cam is fixed, one-half, *h f g*, of the cam is semicircle about the centre *a*, so that the end of a lever resting on the periphery at *g* remains at rest during half a revolution

(in the direction indicated by the arrow), but so soon as the point *h* has passed the end of the lever the latter begins to rise and is uniformly lifted until it reaches the point *c*, with which the point *b* of the cam will then coincide; during the next quarter-revolution the lever will gradually fall to the position *g*. The cam is thus set out: let *f d* be the amount of travel to be given by the cam; from *a* with the radius *a f* describe the semicircle *g f h*, and from the same centre with the radius *a d* describe the circle *b c d e;* draw the diameter *b d* at right angles to the diameter *c g h e;* divide the semicircle *c b e* into, say, eighteen parts, and divide *f d* into nine parts, then, beginning from *f* and with radii corresponding to the divisions on *f d*, describe the arcs shown. Thus, with the first radius beyond *a f*, draw the arcs between the radii *c a* and *i c*, and *e a* and *k a;* the next arcs will be carried to the next radii, and so on, and the points where the arcs meet the radii will determine the periphery of the cam.

Fig. 40.



In machine tools, such as planing, shaping, and slotting machines, a slow movement is required for the forward motion when the cut is on, and a quick back motion to save time between the cuts; this may be effected by using elliptical-toothed wheels so arranged that on the back stroke the long radius vector of the driving wheel corre-

sponds to the short radius vector of the driven wheel; and by different arrangements of segmental wheels; but these cannot properly be classed among cams.

It will be evident on a little consideration that almost any conceivable mechanical motion may be obtained by a judicious use of cams, and, in fact, the different kinds are almost innumerable as used in sewing machines, pin making, and other machinery requiring special and intermittent movements.

CHAPTER XVIII.

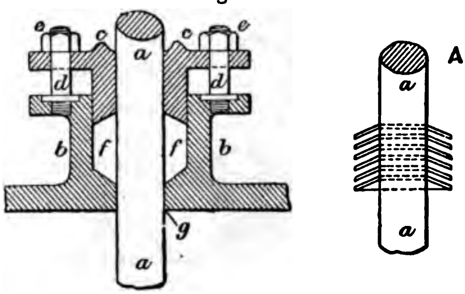
PACKINGS.

The arrangement of packings for pistons and piston rods, and pump plungers and rods, has always been a sort of vexed question, and the difficulty attendant upon the achievement of success is sufficiently evidenced by the immense number of patent packings which have from time to time appeared in the market.

The old-fashioned hemp or gaskin packing is as satisfactory for piston rods as any, but it requires frequent renewal, as, of course, it is constantly wearing away under the influence of friction and heat, and has to be kept up tight against the rod by an adjustable gland of the form shown at Fig. 41, which is a vertical section of an ordinary stuffing box. *a a* is the rod to be packed, *b b* a stuffing box cast on the cover through which the rod *a a* is required to work air or steam tight; in *b b* is a cavity so that when the rod *a a* is in place there is an annular space, *f f*, around it to receive the packing. To the box *b b* is fitted a gland *c c*, having a hole bored through it to fit freely the rod *a a*, which also fits a hole bored at the bottom of the stuffing box at *g*. Into the flanges of the stuffing box are screwed two or more studs *d d*, and holes are drilled in the flange of the gland corresponding

in position to these studs, which are screwed at the ends to receive nuts. The stuffing being put into the space *f f* the gland *c c* is adjusted upon it and tightened up by means of the nuts *e e*, and from time to time as the packing wears away the nuts are further tightened until it becomes necessary to renew the packing material alto-

Fig. 41.



gether. Of course the gaskin must be thoroughly soaked in melted tallow before being applied in the stuffing box. Several kinds of metallic packings for stuffing boxes have been brought out to replace the gaskin; one form is shown at A. This consists of a series of conical rings of metal surrounding the piston-rod, against which they are caused to bear by screwing down the gland upon them, and so to a certain extent flattening them out.

India-rubber packings have been tried, but, as far as our experience has gone, without satisfactory results, there being a tendency to cut the rod *a a* in grooves.

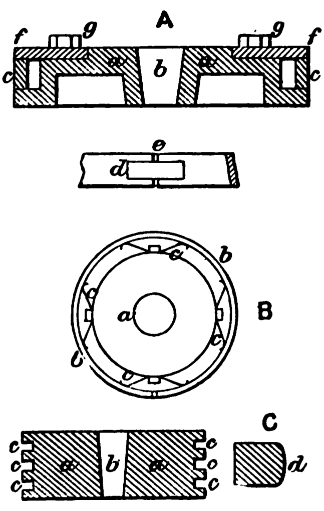
We have recently had our attention drawn to an engine of American design in which no solid packing is used around the piston-rod, which passes through a *long* tube as an easy fit, and *it is said* that there is no leakage of steam.

L 2

If this be true the actual packing must be the water formed by condensation of the steam within the tube, which is retained by capillary attraction with sufficient force to resist its expulsion by the pressure of the steam within the cylinder. We shall not discuss this contrivance, but only give this description of it for our readers to take for what it is worth.

When the rods are very small, the gland *c e*, instead of being forced up by nuts, may be made to screw into the stuffing box *b b*, in which case the flange of the gland will be made of hexagon form like a nut.

Fig. 42.



The piston packings are invariably made of metal, and are commonly arranged as shown at A, B, and C, Fig. 42.

At A is shown a vertical section of a solid-bodied piston, packed with a split cast-iron ring: *a a* is the body of the piston, in which is bored a tapered hole, *b*, to receive the end of the piston-rod; c c^1 shows the packing ring, which is made thinner at c^1 than at *c*. This ring is cut through at one side, as shown at *e*, the cut being made at the thinnest part, and the passage of steam through such cut being prevented by a closely fitting tongue, *d*, fastened by the end to the ring, the other end being free

to allow the elasticity of the ring to come into play. In some cases two rings, one above the other, are used, the joints being placed diametrically opposite each other; *f f* is a junk ring to retain the packing-ring in its place, and is itself secured by bolts *g g*, tapped into the body *a a* of the piston.

The packing ring *c c* is turned of somewhat greater diameter than the cylinder, a sufficient portion is then cut out at *e* to allow of its compression, so that it can be forced into the cylinder, against the interior of which it will then press by virtue of its elasticity; these spring rings are made of cast iron. These rings are, however, only suitable for pistons of small diameter; the object in making them thick on one side and gradually thinner each way towards the slit *a*, is to obtain something approaching to uniformity of pressure on the interior surface of the cylinder.

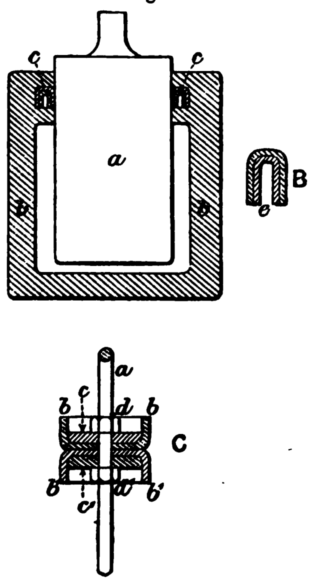
When the piston is of a diameter too large for the spring ring arrangement, a thin ring, or a ring made in segments, may be used, being pressed against the interior of the cylinder by springs as shown at view B, where *a* is the body of the piston, *b b* the packing ring, and *c c c c* the springs by which it is pressed against the cylinder.

At C is shown another method of packing small pistons. The body *a a* of the piston (which also has a tapered hole *b*, to receive the end of the piston-rod) has a number of grooves *c c c* turned on its periphery, into which are slipped split wrought-iron rings of the sectional form shown at *d*. These rings are very narrow, varying from one quarter inch to one inch. The top and bottom edges are rounded off in order to obviate the possibility of their cutting into the cylinder.

We will now proceed to consider the packings to be used for pumps, &c.

The rods of air-pumps, and also their pistons, may be packed in the same way as the corresponding parts of steam engines, but the parts exposed to water should, where the expense is not a positive bar to it, be made of brass, or of iron lined with brass or Muntz's metal. In plunger-pumps and hydrostatic presses it has been usual to pack with leather collars, as shown in Fig. 43. *a* is a plunger or ram, working in a cylinder, shown in vertical section at *b b*. In the upper part of the interior at *b b* is cut a groove to receive the leather ring *c c*, which is more clearly shown in section at B. It will be seen that on the application of pressure to the liquid in the cylinder, a portion of it passing into the hollow collar B will distend it, and so press it against the plunger or ram *a*.

Fig. 43.



Within the collar B is placed a copper ring *e*, to prevent the collapse of the leather when not in use. As we have previously observed, we prefer to use gaskin packings.

At C is shown a pump bucket packed with leather. *a* is the pump-rod, *b b* and b^1 b^1 are two cup-shaped leathers *through* which the pump-rod is passed; the leathers are

secured to the pump-rod by washers $c\,c^1$, held in position by nuts $d\,d^1$. The washers $c\,c^1$ are made of the proper diameter to keep the cup-leathers up to the diameter of the pump barrel, against the sides of which the water pressure acting inside, the leathers will assist in pressing them.

CHAPTER XIX.

BOLTS, NUTS, STUDS, AND SCREWED ENDS.

The whole security of a machine, assuming all its elements to have been properly designed, must ultimately depend upon the bolts, rivets, or other equivalents by which its component parts are held together, hence the great importance of carefully determining their dimensions and securing both good materials and good workmanship. There is, in fact, so much slovenly and dishonest work turned out in the shape of bolts and nuts that we cannot keep too sharp an eye on this part of the work. We unhesitatingly say that all bolts should be *chased* by a cutting tool and not *worked up by dies;* we say "worked up" because die-cut screws are partly cut and partly squeezed, and it is evident that a thread squeezed up cold cannot possibly possess the strength of the material when properly worked.

The strength of a bolt or any description of screwed end must be determined from its diameter measured at the bottom of the thread. In machinery in this country the Whitworth pattern of screws is universally used; the threads are cut, with an angle of fifty-five degrees, and for different diameters the numbers of threads per inch for *angular threads* are given in the following table :—

Diam. inches.	No. of Threads per inch.	Diam. inches.	No. of Threads per inch.	Diam. inches.	No. of Threads per inch.
$\frac{3}{16}$	24	$1\frac{3}{8}$	6	$3\frac{3}{4}$	3
$\frac{1}{4}$	20	$1\frac{1}{2}$	6	4	3
$\frac{5}{16}$	18	$1\frac{5}{8}$	5	$4\frac{1}{4}$	$2\frac{7}{8}$
$\frac{3}{8}$	16	$1\frac{3}{4}$	5	$4\frac{1}{2}$	$2\frac{7}{8}$
$\frac{7}{16}$	14	$1\frac{7}{8}$	$4\frac{1}{2}$	$4\frac{3}{4}$	$2\frac{3}{4}$
$\frac{1}{2}$	12	2	$4\frac{1}{2}$	5	$2\frac{3}{4}$
$\frac{5}{8}$	11	$2\frac{1}{4}$	4	$5\frac{1}{4}$	$2\frac{5}{8}$
$\frac{3}{4}$	10	$2\frac{1}{2}$	4	$5\frac{1}{2}$	$2\frac{5}{8}$
$\frac{7}{8}$	9	$2\frac{3}{4}$	$3\frac{1}{2}$	$5\frac{3}{4}$	$2\frac{1}{2}$
1	8	3	$3\frac{1}{2}$	6	$2\frac{1}{2}$
$1\frac{1}{8}$	7	$3\frac{1}{4}$	$3\frac{1}{4}$		
$1\frac{1}{4}$	7	$3\frac{1}{2}$	$3\frac{1}{4}$		

Square-threaded screws will have half the number of threads per inch that are found on angular or V-threaded screws.

The depth of thread is equal to the pitch, and one-quarter of the depth is rounded.

Some mechanics will allow five tons as the safe working tensile strain per sectional square inch, but we do not think it advisable to assume more than 9,000 lbs., and on that factor the following table of strength of bolts is calculated. Column D is the diameter of bolt in inches, and column W the working load in lbs. either in tension or shearing strain, the latter of course being taken for single shear through the bottom of the thread :—

D.	W.	D.	W.	D.	W.	D.	W.
$\frac{3}{16}$	77	1	3972	$2\frac{1}{4}$	21640	$4\frac{1}{2}$	102764
$\frac{1}{4}$	162	$1\frac{1}{8}$	5045	$2\frac{1}{2}$	28270	$4\frac{3}{4}$	114117
$\frac{5}{16}$	286	$1\frac{1}{4}$	6510	$2\frac{3}{4}$	33583	5	128648
$\frac{3}{8}$	440	$1\frac{3}{8}$	7713	3	44620	$5\frac{1}{4}$	142372
$\frac{7}{16}$	600	$1\frac{1}{2}$	9470	$3\frac{1}{4}$	49556	$5\frac{1}{2}$	158596
$\frac{1}{2}$	784	$1\frac{5}{8}$	10610	$3\frac{1}{2}$	59263	$5\frac{3}{4}$	173506
$\frac{5}{8}$	1400	$1\frac{3}{4}$	12873	$3\frac{3}{4}$	67448	6	190849
$\frac{3}{4}$	2135	$1\frac{7}{8}$	14525	4	77752		
$\frac{7}{8}$	3014	2	16961	$4\frac{1}{4}$	89371		

The thickness of the head of the bolt should be equal to the diameter of the bolt, and its width over the angles equal to twice the diameter of the bolt, in order to insure ample strength against stripping of the thread. Theoretically the head does not require to be so thick, for as the tensile and shearing strengths per square inch are the same, the shearing areas should only require to be the same.

Now the sectional area of the body of the rivet will be found by multiplying its diameter squared by ·7854. If the head shears off it will be by the body, as it were, drawing out of it; hence the area in this case will be the circumference of the body of the bolt (that is, the diameter multiplied by 3·1416) multiplied by the height or thickness of the head. These two areas, then, are required (for perfect material and workmanship) to be equal. Let us take a case to find the required proportion of height to diameter, calling the latter one inch:—

The cross-sectional area of the bolt will be ·7854 square inches; the stripping area of the head will be 3·1416 multiplied by the height, so that the theoretical height will be—

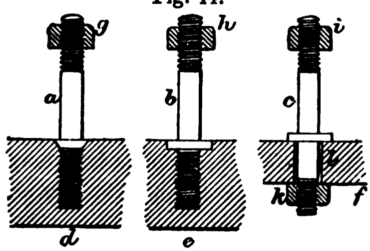
```
3·1416)·78540( ·25 inches.
        62832
        ------
        157080
        157080
        ------
```

or the head must be at least ¼ the diameter of the bolt in thickness. We have, however, to consider that the shearing strain on the head acts *parallel* to the grain of the iron instead of across it; hence the resistance will not be *so great*, so that in fact, all things considered, it is better

to adhere to the practice of making the thickness of both heads and nuts equal to the diameter of the bolt. The theoretical thickness of the nut would be half a diameter on account of the loss due to rounding out the bottoms of the threads.

The same proportions will apply to screwed ends of distance rods, &c. A few words are appropriate here in connection with studs and dead centres. The stud may consist merely of a rod screwed at both ends, one being screwed into the framework, and the other fitted with a nut for securing any required part of the machine on the stud; but studs with collars are decidedly preferable, as they can then be screwed up to a fixed distance with more certainty. In Fig. 44 *a* is a stud with a collar fitting into a countersink in the frame *d*, *g* is the top nut; *b* is a stud with a cylindrical collar, which, however, probably, does not steady the stud so completely as the conical form, *h* is the top nut, and *e* the frame into which the stud is screwed.

Fig. 44.



c is a dead centre passing through the frame *f*, and secured beneath by a nut *k*; the part of the dead centre immediately below the collar has a rib, or feather, *l*, which fitting into a groove in the frame *f*, prevents it from turning, which might otherwise happen if the rocking or revolving element supported by it has a very tight fit. The top or outer nut *i* screws up to a certain point, where it jams against the metal at the end of the screw

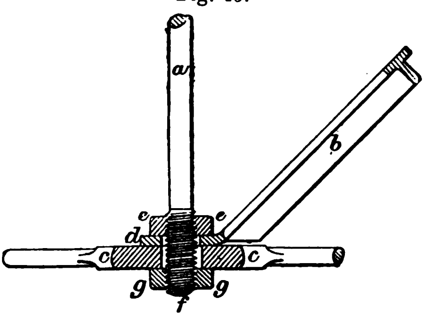
thread, which is carried far enough to leave just the proper space between the nut and the collar for the revolving or rocking element to work freely.

For the practical method of chasing screws see the author's "Practical Treatise on Mechanical Engineering."

There is among ignorant labourers a great tendency to overstrain the screws in tightening up nuts, and also screws that are tapped into framework. It is quite sufficient to tighten up the nut or screw so that it takes a fair solid bearing, for if it is further strained the only effect obtained is to strain the threads, and make them what is called " drunk," when after a few removals and replacements of the nut the thread becomes worthless.

In classes of machinery where there is much vibration, the nuts having, in the first instance, been screwed up to their proper bearings, a hole may be drilled right through the nut and bolt, and a steady pin put in, and this steady pin, which is slightly tapered, should be split at the narrow end, so that it may be slightly spread after being driven into its place.

Fig. 45.



At Fig. 45 is shown an example of a screwed end, commonly used in roof joints. *a* is a queen, or minor rod; *b*, a T-iron strut, having the web cut away at the lower part, and the table (shown in *section*) at *d* bent so as to lie flush upon the main tie *c c*.

e e is one, and *g g* another nut on the screwed end of the rod *a*, and between these nuts the strut *b* and the tie *c c* are secured to the rod *a*.

The screwed end is made of a forging sufficiently large to have a diameter *at the bottom of the thread* slightly in excess of that of the rod *a*, so that the strength of the latter may be fully equalled by that of the screwed end, which is welded on to it.

It will be seen that the holes in the table *d* of the strut *b*, and in the square part of the main tie *c c*, are made of a diameter large enough to allow the screw to clear. As when the roof principal is once adjusted there is no reason for altering the nuts, the end *f* of the rod *a* may be cold riveted over the nut *g g*.

CHAPTER XX.

STANDARDS AND FRAMEWORK.

In designing standards and framework of every description the first desideratum is solidity and rigidity, and this should be attained as much by form as by mass of material.

The form, then, of the parts of frames that act as bracings must be determined with due regard to the nature and direction of the strains to which they will be subjected.

Take, for instance, an ordinary horizontal steam engine; supposing the work to be properly balanced and truly adjusted, we have to provide against alternate longitudinal tension and compression simply, but we must form the frame so as to hold the plummer blocks in suitable positions, so that the directions in which they can be adjusted are in line with the maximum strains. A faulty and a correct method of arranging for the bearings are shown at Fig. 46, these being the bearings for the main shaft.

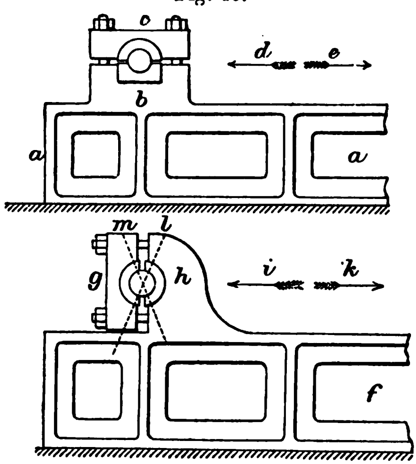
a a is one frame carrying a plummer block *b*, fitted with brasses and a cap *c*; now here it is evident the adjustment is arranged vertically, whereas the strains are horizontal in the directions of *d* and *e*, and no amount of tightening *up the brasses* by screwing down the cap, will in any way

compensate for the wear due to the longitudinal strains, though it will compensate for the extra wear due to the weight of the main shaft, fly-wheel, &c., which acts vertically.

f f shows another form of bed, with the plummer blocks more suitably arranged.

We have seen that, in addition to the wear caused by the mere friction of the shaft revolving in its bearings,

Fig. 46.



there are two directions in which extra or special strains occur—that is, horizontally and vertically. The latter is chiefly due to the weight of the fly-wheel, driving drums, &c., the main shaft itself having an effect comparatively insignificant; if, then, separate plummer blocks are provided for the shaft close to the fly-wheel, &c., arranged as shown at *c b;* in dealing with the bearings close to the

crank we should arrange them as shown by the full lines at *h* the plummer block, and *g* the cap; then will the adjustment line in the direction of the strains, as shown by the arrows *i* and *k*.

At first sight it might appear to the inexperienced that instead of using a *vertical* line of division for the brasses, that an *inclined* one would meet the cases of both extra strains; but this is not so, as when the strain from the piston is acting in the direction of the arrow *i*, the line of division would be indicated by the position of some such line as that shown dotted at *l*, and when the strain acts in the direction of the arrow *k*, the line of division should be located as shown by the dotted line *m*, and as the line cannot be varied with each stroke of the piston, the only thing to do is to keep to the vertical division, as shown by the plummer block and cap *h*, *g*. In a vertical engine, the piston strains and the weight acting both in a vertical direction, the adjustment is correct when the line of division of the bearings is horizontal.

The tensile strain on the bolts will be equal to the maximum strain on the piston, which is found by the following:—

RULE.—*Multiply the square of the diameter of the piston, in inches, by* ·7854, *and by the maximum boiler pressure in lbs. per square inch. The product divided by the number of bolts proposed to be used will give the strain in lbs. upon each bolt.*

For example, let there be an engine working on a single crank, and therefore acting on *one* plummer block and bearing, and let the cylinder be 24 inches in diameter and the maximum boiler pressure 65 lbs. per square inch, *it being* proposed to use *four* stud bolts to secure the cap, *then:*—

24 inches diameter of piston

24 " "

96

48

576 square inches area of piston (or cylinder)

65 lbs. per square inch, maximum boiler pressure

2880

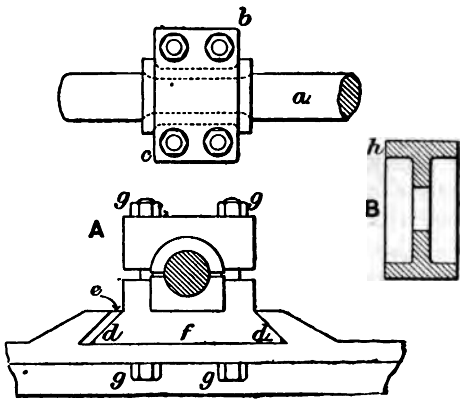
3456

4)37440 total pressure on piston

9360 lbs. strain on each bolt.

A reference to the table of working strengths for bolts shows that the nearest in strength above that required is

Fig 47.



a 1½ inch bolt, which should therefore be adopted. The bearing in plan will appear as shown at Fig. 47; a is the

end of the main shaft intended to receive the crank, and *b c* the cap of the plummer block.

The plummer blocks should in every case where it is practicable be cast in one piece with the bed frame of the engine, in order to avoid the introduction of another joint; but where it cannot be avoided a pedestal of the form shown at A may be used. This bearing is arranged for extra vertical strains. The plummer block is made with bevelled flanges *d d*, which fit into undercut grooves as shown, the whole being tightened up by a wedge *e*. The cap is fastened to the plummer block *f* by bolts *g g*, which for greater security may be taken right through the flange of the bed frame as shown. The cast-iron frame will generally be of the section shown at B, and as cast iron in tension will break at about 8 tons or 17,920 lbs. per sectional square inch it is not safe to allow for working strain more than 1,800 lbs. per square inch. In the case considered above we shall therefore require a sectional area, to be found by the following general

RULE.—*To find the sectional area in square inches of the top flange (h,* B, *Fig.* 47) *of a cast-iron frame, divide the maximum gross pressure in pounds upon the piston by* 1800.

```
1800)9360 lbs.(5·2 square inches sectional
     9000           area of top flange.
     ────
      3600
      3600
      ════
```

If, therefore, the top flange be made six inches wide and three inches deep and five-eighths inch thick, the requisite area will be given, but this is exclusive of bolt holes; hence it must be widened where those occur, as, for instance, under the plummer block, where, to admit

two 1¼ inch bolts in its width, the flange itself must be increased to nine inches in width; it must also be similarly widened when bolts for the attachment of the cylinder and guide bars occur.

In connection with framing we may also refer to foundations, for it is evident that a solid foundation is necessary to support the superincumbent load, and it is also necessary that the machine should be sufficiently firmly anchored to the foundation to prevent rocking or jumping.

In order so to fix the bed plate, anchor bolts must be used passing down through the foundation, and passing through an anchor plate at the bottom, beneath which they are held by their heads. The anchor bolts are placed head downwards in the excavation for the foundation, and the anchor plates being passed down over the bolts until they rest upon the heads the foundations are built up and the bed plate of the engine secured on the anchor bolts, which pass through them by means of ordinary nuts. For light engines on good substantial bed plates it is quite sufficient to fasten them to their foundations by short rag bolts run in with cement or sulphur to prevent lateral movement from vibration.

Some classes of very large engines cannot be carried by a continuous bed plate or framework and require special foundations in order to insure their stability, and we cannot for this case take a more striking example than that which occurs in the single-acting Cornish engine. In this machine the piston in its down stroke lifts a "preponderating weight" hung to the far end of the main beam, and on the escape of the steam from the cylinder the preponderating weight descends, actuating the pump plunger, and also drawing the steam piston again to the top of the cylinder. Now it is evident that if the cylinder

be not held down with sufficient force that instead of raising the preponderating weight, the steam acting on the inside of the cylinder will lift the latter and slide it up the piston-rod. The cylinder, then, must be anchored down to a sufficient mass of masonry to obviate this, and the weight of that mass should not be *less* than one and a quarter times the preponderating weight; it is called the "cylinder load." The horizontal length and breadth of the cylinder load being given in feet, also the weight of its materials in pounds per cubic foot, its required depth will be found from the following:—

Rule.—*To find the depth in feet of the cylinder load multiply the length in feet by the breadth in feet and by the weight of the material in pounds per cubic foot; by the product divide one and a quarter times the preponderating weight in pounds; the quotient will be the required depth.*

Let the preponderating weight be *95,000 lbs. and the cylinder load 12 feet long and 10 feet broad, and built of stone weighing 135 lbs. per cubic foot.

```
      12 feet length of cylinder load
      10  ,,  breadth        ,,
      ——
     120
     135 lbs. per cubic foot of stone
    ————
     600
    360
   120
   —————
   16200
    1·25
   —————
   81000
  32200
 16200
————————
20230·00
```

```
20230·00)*95000 lbs.(4·696 feet nearly, say, 4 feet
         80920              8½ inches depth of
        ------              cylinder load.
         140800
         121380
         ------
          194200
          182070
          ------
           121300
           121380
           ------
```

Care must be taken that the stones composing the cylinder load are well cemented and bonded together so as to form a concrete mass, so that it shall not disintegrate when the strain comes upon it; the anchor plates must also be of large area, or it is better where possible to make them continuous.

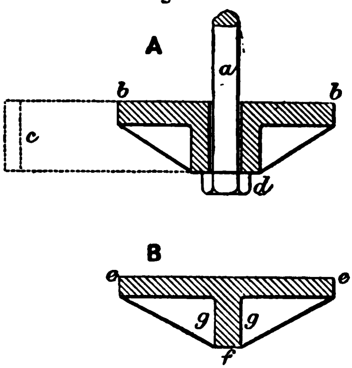
The total load being *95,000 lbs., we can find the number of bolts of a given size required by dividing that weight by the working strength of one bolt, as shown in our table (page 225). Let 1½ bolts be decided upon, then each such bolt has a working strength of 9,470 lbs.

```
Strength of 1½ bolts in lbs. 9470)*95000(10 bolts,
                                   9470
                                   ----
                                     300
                                     ...
```

or rather, as there is a remainder, eleven bolts will be required, but for the sake of symmetry we should in practice use twelve bolts. We may have the anchor plate at the bottom of the cylinder load in the form of a ring of the section shown at Fig. 48, where the bolts occur, at A, and in the parts intermediate between the bolts as shown at B.

a is the holding-down or anchor bolt, of which *d* is the head, *b b* is the anchor ring, having cast on it bosses for the heads of the anchor bolts to bear against. These bosses

Fig. 48.



are stiffened transversely by brackets as shown, and longitudinally by a rib running from boss to boss, shown in section at *f*, view B, where *e e* is the top of the ring, in this part of which brackets *g g* are also cast at intervals.

When the cylinder load is very wide it may be advisable to have radial ribs, as shown by the dotted lines at *c*, crossing the interior of the ring to support the masonry, the ribs, of course, meeting in a boss in the centre. This arrangement is most called for when the cylinder load is of brickwork. A general rule for the strength of unflanged ribs, such as *c*, may here be given, which rule may be transposed as circumstances may require.

Rule.—*To find the safe load on a rectangular rib in pounds equally distributed over its length: all dimensions being taken in inches*, multiply the breadth *by the square of*

the depth and by 9677, *and divide the product by the bearing on clear span.*

Let a rib be 8 feet (96 inches) in length, 6 inches deep, and 2 inches thick; required the safe working load.

6 inches depth
6 „
36 square of depth
2 inches breadth
72
9677 constant
504
504
432
648
Bearing 96)696744(7257 lbs. load.
672
247
192
554
480
744
672
72
..

RULE.—*The load in pounds and the depth and span being given to find the thickness, multiply the load by the span and divide the product by* 9677 *times the square of the depth.*

Let the load be 7,257 lbs. equally distributed, the depth 6 inches, and the span 96 inches. Then :—

```
          9677 constant
            36 square of span
         ------
         58062
        29031
        ------
        348372*
        =======
     7257 lbs. load on rib
       96 inches span
    ------
     43542
    65283
    ------
348372)696372(1·99, &c., inches thickness of rib.
    348372
    -------
    3480000
    3135348
    -------
     3446320
     3135348
    -------
      310972
       . . . .
      =====
```

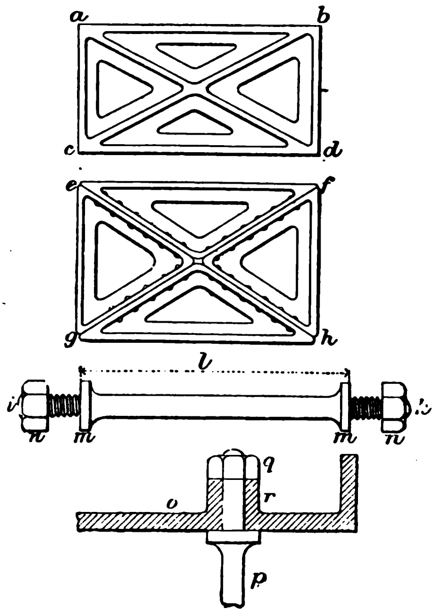
This should prove the previous example, and the slight discrepancy is due to dropping the decimals in the former case, which showed a remainder. Of course in practice 1·99, &c., would be called 2 inches.

In no case does the necessity of a self-contained framework appear more imperative than in that of marine engines, for even in locomotive work the engines and

crank shaft are not rigidly preserved in certain relative positions, inasmuch as a certain amount of play is allowed by the springs.

It is evident that no strain should be permitted to come upon the framework or skin of the ship from the vibration of the engines, or from the reactions arising among their elements, hence rigidly braced frames must be employed. Frames of a similar type must also be adopted for various classes of machines where the lines of strain of the different parts cross each other; we may, therefore, deal generally with cross-braced frames.

Fig. 49.



In Fig. 49 *a b c d* represents a typical cast-iron-braced frame; it is stiffened by the diagonal ribs *a d* and *b c*, which resist any force tending to disturb its rectangular form; this frame is shown as being all cast in one piece. Its connection with the other framed sides of a machine will be made through suitable lugs and

bolts, and, where necessary, distance rods of the form shown at *i k*.

The exact distance required between the frames having been determined, the distance rods are formed with collars *m m* to correspond, making the distance *l* equal to that required. The ends *i* and *k* having been passed through opposite side frames, the latter are held up to position by the bolts *n n*. Even where four braced side frames are used it is advisable to employ tie rods passing right through; but in this case the collars *m m* will be omitted, as the castings will abut against each other, and so preserve the proper distance, while we have the advantage of the superior tensile strength of wrought iron in the tie.

In all cases where a bolt or tie passes through a cast-iron plate, the metal should be thickened up to a circular boss, as shown in vertical section at *r*. *o* is the ordinary thickness of the plate, *p* is part of the distance rod, and *q* the nut.

e f g h shows another kind of braced frame made of wrought angle iron. The angle irons are bent up into four triangles, and welded up; they are then riveted together, as shown, and in this form make a very strong frame.

In bending angle irons to such shapes the positions of the corners are marked on the straight bars, and V-shaped pieces there cut out of the vertical limbs; the angle irons are then heated and bent to the required forms, and the free ends, and also the edges of the Vs, securely welded together.

A question naturally arises as to which kind of frame is the better—a cast or a wrought iron one—and there is something to be said on each side.

In the first place as to cost: there is, probably, not

much difference, for although the wrought-iron frame will certainly cost more per ton, yet, as it will be much lighter, the additional rate will be compensated for to some degree, and perhaps altogether.

The wrought-iron frame has the advantage of superior general strength, of more uniform quality, and an exemption from liability to fracture, which allows us to use much lighter sections than can safely be employed in cast iron; but, on the other hand, all bearings and plummer blocks must be bolted or riveted to the wrought-iron frame, whereas if cast iron is the material used, they can be cast on so as to form one piece with it. Furthermore, if *weight* is required in order to give steadiness to the machine, then cast iron has the advantage in this point.

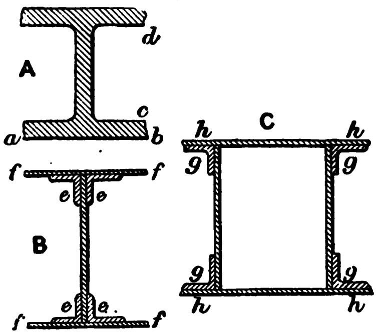
As we cannot determine in actual measured strain the force acting upon the cross bracings, the determination of the sizes of those parts becomes a matter of practice, rather than of calculation.

It often happens that girders are required to sustain the framework of machines, or for the attachment of tackle for taking them to pieces for cleaning or repairs; hence it is necessary to give rules for calculating the sizes of such simple forms as may be called for.

There are two principal ways in which the girders may be loaded—1st, with a load at the centre; 2nd, with a load equally distributed over the clear span. In the latter case a given girder will carry twice as much as in the former, and for this latter case we shall give rules, it being understood that the same rules may be applied for the central load; but for purposes of calculation that *central load must be doubled.* In Fig. 50 A is a section of a cast-iron girder; *a b* is the bottom flange, *c d* between the flanges is the depth. The web connecting the flanges

vertically should be of a thickness not much less than that of the flanges. B is a single-webbed wrought-iron girder, the flanges *f f*, *f f* being joined to the web by angle irons *e ee*. The web should be ⅜ inch thick for all such sizes as are likely to occur in the cases we are dealing

Fig. 50.



with, and if the girder exceeds a foot in depth vertical, T-iron stiffeners, 6 inches by 3 inches by ½ inch, should be placed on both sides of the web every 4 feet, and riveted through.

C is a double-webbed girder, similarly connected by angle irons; in this case the webs may each be ¼ inch thick, and stiffened each web on *one* side by T-irons of the dimensions and spacing given for single-webbed girders.

Rule.—*To find the sectional area of one flange of a cast-iron girder in square inches, under a uniformly distributed load, multiply the load in tons by the span in feet, and divide by the depth in inches.*

Example.—Let the load be 8½ tons, the clear span of the girder 13 feet, and its depth between flanges 11 inches.

8·5 tons distributed load
13 feet clear span

255
85

Depth in inches 11)110·50

10·04 square inches of sectional area of flange.

The flanges in this case would be 10 inches wide and 1 inch thick, making the total depth of the girder over the flanges 13 inches. *Generally* cast-iron girders are made with a smaller top flange in the ratio of 4 to 1, but for cases such as those with which we are now dealing, the broad top flange is generally required to give a good bearing.

To find the sectional area of one flange in either B or C we have the following:—

RULE.—*Multiply the distributed load in tons by the span in feet, and by ·375, and divide the product by the depth in inches; the quotient will be the sectional area of one flange in square inches.*

EXAMPLE.—Let the load be 11½ tons, the span 12 feet 6 inches, and the depth between flange plates 14 inches.

11·5 tons distributed load
12·5 feet clear span

575
230
115

Depth in inches 14 { 2)143·75
7) 71·875

10·26 square inches sectional area of flange.

This sectional area will be made up by the flange plate and by the horizontal limbs of the angle irons; if these latter, the angle irons, are of the (for small size girders) usual size, 3 inches by 3 inches by ½ inch, the two horizontal limbs will together make up three sectional square inches, which, deducted from 10·26 square inches, leaves 7·26 square inches to make up in the flange plate; if, then, we use a plate a foot wide and ⅝ inch thick we shall have, as ⅝ is equal to ·625,

·625 inches thick
12 inches wide
7·500 sectional square inches.

An amount slightly in excess of that calculated.

Without far exceeding the space at our disposal, we cannot further exemplify the different forms of framework, but the remarks we have made should be sufficient to guide the practical man or student to the right course to pursue in any case.

There are several points always cropping up which one would imagine common sense would settle; one is that a heavy wheel should not be put *outside* a bearing if it can be avoided, but an instance came under our notice some years since where the fly-wheel of a factory engine being on the outer end of the main shaft, came off while the engine was running at full speed, the result being, as may be imagined, that the wheel rolled through the works destroying everything in its course until its momentum was expended.

As a general rule also, a bed of good concrete (1 part barrow lime to 5 parts clean gravel) is more reliable for a foundation than one built up of stone, for if any of the *stones be laid* on a *hollow bed* they are almost certain to crack.

CHAPTER XXI.

COUNTERBALANCES AND THEIR EQUIVALENTS.

THE inconvenience arising from the vibration and sudden shocks arising in machines having heavy elements in reciprocating motion, and from the disturbance ensuing from the rotation of excentric masses, has for many years occupied the attention of some of our most skilful and painstaking mechanicians, but we cannot, even now, say that a perfect remedy has ever been approached.

Take, for instance, one of the most familiar cases, that of the locomotive engine, and consider the disturbing influences called into play when it is working.

In the first place, the steam entering the cylinder presses equally upon the piston *and upon the cylinder cover* and, as the frame of the locomotive is not fixed on a foundation, a certain rebound must occur, its extent depending in great measure upon the weight of the engines and boiler; this rebound will change in direction at each stroke of the piston, and, as one piston is a quarter of a revolution, or half a stroke, in advance of the other, the result of all the rebounds will be to produce a sort of wriggling motion in the frame of the engine. Next, we have the sudden reversal of the direction of motion in the heavy reciprocating masses, and the swing of the cranks

and excentrics; this latter produces a vertical movement, the first disturbance being lateral.

Now it is very evident that these various forces acting, we may say, in all directions, must produce most destructive strains upon the framework, as well as on the working parts of such machines as are subjected to them; hence the necessity of devising some method of, as far as possible, eliminating such injurious effects.

There is, moreover, a loss of efficiency in suddenly changing the direction of motion of a mass reciprocating in a straight line, as at each change the whole of the momentum of the mass must be absorbed before its motion can be reversed, and this absorption will generally be at the expense of the various bearings, so that the energy represented by that momentum is not only lost but has actually a destructive effect.

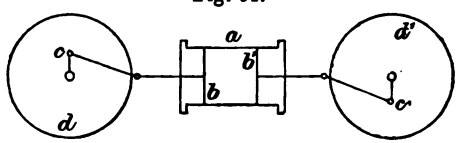
The reaction of the steam against the cylinder cover has never been satisfactorily eliminated, though some years back Mr. Bodmer endeavoured to prevent any external disturbance by using two pistons in one cylinder: the upper piston-rod was tubular, the lower one passing through it. In this contrivance the pistons would at one end of the stroke of the engine be together in the centre of the cylinder, and at the other end of the stroke one piston would be at each end of the cylinder. The mode of action was as follows: Assuming at the start that the pistons are together, the steam is admitted between them, forcing one up and the other down, and thus producing no pressure on the cylinder ends; on the return stroke the steam is admitted above the top piston and below the bottom one, and they are again brought together; and at the same time the steam *acts equally* upon the top and *bottom covers* of the cylinder, and therefore, acting equally

in both directions, does not tend to displace the cylinder. The engine, however, so constructed, is too cumbersome and costly to gain favour, and therefore has dropped out of sight, but the method is worth attention, as it may be applicable to some descriptions of machinery.

There does not seem any reason why a properly designed two-piston cylinder arrangement should not be adapted to the locomotive engine, in order to reduce, or altogether obviate, the wriggling movement alluded to on a previous page.

In this case the general disposition of the parts would be as shown in diagram form at Fig. 51.

Fig. 51.



a is the steam cylinder, in which work the two pistons $b\ b^1$, running in opposite directions, and acting on cranks $c\ c^1$, on the shafts of which are fixed the coupled driving-wheels $d\ d^1$. In this arrangement it is evident that no longitudinal oscillation can come upon the frame from the steam cylinder, as when the steam is acting *between* the pistons $b\ b^1$ there is no pressure on the cylinder ends, and when the steam is acting *outside* the pistons the steam pressure is the same on *both ends* of the cylinder; hence there can be no tendency to disturb it.

We will now pass on to the equilibrating of cranks and other excentrically disposed revolving masses.

In regard to a crank, considered by itself, it may be counterpoised by placing a similar weight symmetrically on the opposite side of the centre ; or, as is often done in the case of engines of moderate size, the crank may be

replaced by a circular disc, in which the crank pin is fixed, and which will, of course, run equably upon its bearings. This, however, though getting rid of the crank disturbance, does not compensate for that of the connecting-rod, of which half the weight (more or less according to its form) rests upon the crank pin. This weight having been accurately determined, should be counterpoised by one producing equal effects on the opposite side of the centre of the crank shaft.

If, however, we counterpoise the crank and connecting-rod together by a weight fixed in the fly-wheel of a stationary or the driving-wheel of a locomotive engine, the weight of such counterpoise may be determined in the following manner:—

Determine the position of the centre of gravity of the crank, with its crank pin attached, which may be calculated or ascertained practically by finding the point on which the crank will balance when suspended, the distance of this point from the centre of the crank shaft in inches is to be measured, then *the weight of the crank in lbs., multiplied by the distance of its centre of gravity from the centre of the crank shaft in inches, will give the moment of force due to gravity about the centre of revolution.* This is the moment to be counterpoised, so far as the crank itself is concerned. Now for the connecting-rod. The centre of gravity of this is to be determined in like manner, and the distance of that point from the centre of the piston-rod, in inches, measured, as likewise the length of the connecting-rod in inches from centre to centre of bearings. Then the part of the weight carried by the crank pin will be found by *multiplying the weight of the connecting-rod, in lbs., by the distance of its centre of gravity from the centre of its bearing on the piston-rod head, in inches, and dividing the product by the length, in inches, of the connecting-rod between centres.*

The moment of this force, in inch-lbs., will be the weight on crank pin, multiplied by the radius of the crank.

These two moments are to be added together, and their sum being divided by the distance, in inches, of the centre of gravity of the counterpoise from the centre of the crank shaft, the quotient will be the weight, in lbs., of the counterpoise.

Let us take an example where the radius (of the single crank) is 12 inches, its weight 38 lbs., and its ascertained centre of gravity 5 inches from its centre of rotation. Let the connecting-rod be 4 feet (48 inches) long, between centres, and its weight 77 lbs., its centre of gravity being 28 inches from the centre of the piston-rod head. Then we find the required moments thus:—

38 lbs., weight of crank
5 inches, centre of gravity to centre of shaft

190 inch-lbs., moment of crank.

77 lbs., weight of connecting-rod
28 inches, centre of gravity to centre of head of piston rod

616
154

Length of connecting-rod in inches } 48)2156(*44·91 lbs. on crank pin
192

236
192

440
432

80
48

32

*44·91 lbs. on crank pin
12 inches, radius of crank

8982
4491

538·92 inch-lbs., moment of connecting-rod.

Let the distance of the centre of gravity of the counterpoise be fixed at 25 inches from the centre of the crank shaft.

Distance of centre of gravity of counterpoise from centre of shaft in inches. .	538·92 inch-lbs., moment of connecting-rod 190·00 ,, ,, crank 25)728·92(29·15 lbs., weight of counterpoise 50 228 225 39 25 140 125 15

In coupled locomotives the coupling-rods must also be counterpoised, the calculations being made in a similar manner.

In a four-coupled engine each crank pin will carry half the weight of a coupling-rod, and in a six-coupled engine the crank pin of the centre wheel will carry *half* the whole *length of* coupling-rod, and those on the leading and trailing

wheels will carry *one quarter* each of the whole length of coupling-rod.

The movements of the excentrics are so small in amplitude as not to require any special counterpoise absolutely, though, of course, the more thoroughly an engine is counterbalanced the better will be its performance, and its durability will be also proportionately increased.

Up to this point we have dealt almost exclusively with the counterbalancing of steam engines; but it is evident that those elements treated of will also occur in other classes of machinery under precisely similar conditions, so that the above rules will also apply to them wherever they may occur.

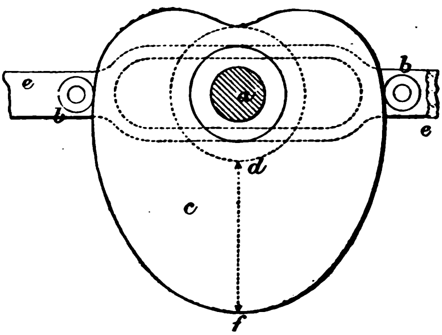
We will now, however, devote our attention especially to machinery generally, and show how to reduce the inconveniences arising from the various cam and other motions constantly coming under our notice.

No doubt it will be immediately objected by many that by counterbalancing the elements of a machine we materially add to its weight, and this is undoubtedly true; but it does not follow that it is not the better course to pursue, for, if by the addition of certain *inexpensive* dead weight, we can insure a smoother running of *any* machine, and so prevent it from knocking itself to pieces, it is certainly true economy to add such weight.

The most troublesome movements to counteract are those of a percussive character, occurring in machines carried in continuous framework. The frame of a steam hammer is, of course, not affected by the concussion, as the anvil is distinct from the standards and has its own special foundation. In small machinery, that is machinery in which the elements are small, edge cams may be used without any counterbalance, as their weights are

quite inconsiderable in proportion to the bulk of the framework, but when the cams are of considerable size they should be counterbalanced, unless instead of using edge cams we can introduce grooved face cams, a method which is analogous to using a disc instead of a crank. We take, for example, a large edge cam of the form shown at Fig. 52. *a* is the cam shaft carrying the cam *c*, which operates on the two rollers *b b*, carried on dead centres fixed in the bar *e e*. This bar, being carried in suitable guides is caused to move longitudinally back and forth at

Fig. 52.



each revolution of the cam *c*, which is so set out that all lines drawn through its centre of rotation and terminated by its perimeter shall be of equal length. The reciprocating bar *e e* is slotted, as shown by the dotted line, in order to allow of its clearing the cam shaft. The amount of travel given to the bar *e e* will be equal to *d f*.

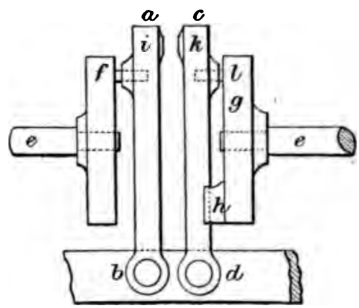
Now it is evident that such a cam unbalanced will, even if revolving at a moderate velocity, give rise to a good deal of chattering, and it is also evident that putting a counterbalance on the other side of the centre of rotation

would be a clumsy arrangement, so we should, where it can conveniently be managed, replace such a cam by a face cam or disc, having a groove of the required shape cut in it.

One friction roller only will then be required; it will be carried on a dead centre on the sliding bar, and will fit so as to run easily in the groove of the face of the disc, which will, of course, rotate with a steady motion.

When cams are used to produce a percussive effect through the medium of dies, by using a double action, the strain may be confined to the cam shaft and so kept clear of the framing; and the advantage of this is that the jarring consequent upon each blow affects no other element of the machine but that to the action of which it is due; this is rather an important matter in machines making two or three hundred revolutions per minute.

Fig. 53.



A suitable arrangement is shown at Fig. 53.

a b is an arm fixed at *b* and furnished with a clip at *i* to hold the work while it is operated on by the die *k*, carried at the end *c* of the arm *c d*, which moves on a dead centre at *d*, being held in the position as illustrated by a spring not shown in the figure. *f* is a plain disc keyed on the cam shaft *e e*, which also carries the face cam *g* (the part of *e e* between *f* and *g* is omitted to show the arms *a b*, *c d*), having a camber *h* which, coming against

the upper part of *k d* at every revolution, drives the die *k* towards the clip *i*, which latter is sustained against the blow by the disc *f*, upon which the friction roller *g* always bears; thus the only effect upon the machine of the strain put upon *k* and *i* is to bring tensile strain upon that part of the cam shaft *e e* which lies between the disc *f* and the cam *g*. *l* is a friction roller behind the die *k*.

We might extend these remarks almost without limit, but having dealt with various typical cases we shall now pass on to the next chapter.

CHAPTER XXII.

REPAIRS AND ADDITIONS TO MACHINERY.

THE repairing branch of the mechanical engineering business is much more extensive than might at the first thought be expected, and it may be incidentally mentioned that it is also very remunerative, as it being almost impossible to estimate the cost of ordinary repairs they are charged for according to the time and material actually expended.

In all descriptions of machinery manufactured for abroad, and especially for the colonies where skilled labour is not obtainable, care should be taken that before it leaves the works the working drawings are examined, and, if necessary, corrected so as to agree exactly with the work *as executed in every detail;* then at any future time, if any part of the machine breaks down or wears out, the makers can prepare a new part to the drawings and dispatch it to its destination perfectly confident that it will properly fit into its place on arrival. Very frequently in repairing old machinery, especially steam engines, endeavours are made to improve its working, but such improvements must not be rashly approached; due consideration must be given after a careful examination of the existing machine.

The great tendency at the present time towards improved

efficiency is by the use of increased and increasing steam pressures and high degrees of expansion; therefore, in introducing these advanced ideas in dealing with old-fashioned machinery we must first make sure that the various parts are of sufficient strength and stability to withstand the increased pressures.

It is almost startling to look back to the days of 7 lbs. per square inch steam pressure and then see *our* locomotives running under 140 lbs. per square inch; but the economy all ways is undoubted, for higher rates of pressure mean reduced capacity of cylinder, and therefore reduced prime cost, and also a saving of space occupied, and a higher degree of expansion means saving in consumption of fuel.

Patching up a boiler is often a very unsatisfactory job, and in the long run it would frequently be much cheaper to have a new boiler at once. An instance came under our notice some years back of a boiler working at a silk mill near Lewisham, where the boiler to be repaired was so deteriorated that on unsetting it and tapping the bottom with a hammer the hammer went *clean through the plate*, showing clearly that at that point it was only the setting that prevented it from bursting.

The safety-valve had no proper weights upon its lever, but from a calculation of the lumps of scrap iron hung upon it, it turned out that the valve lifted at 44 lbs. per square inch. In this case the employés were certainly working at the peril of their lives, for if that boiler had yielded at the bottom the shell would have been hurled upwards, to the destruction of the surrounding buildings.

Recaulking boilers is an operation requiring great care, as there always exists a tendency to split the edges of the *plates*. If we can ascertain certainly what plates are

sound and what are defective, we can cut the latter out and replace them by new plates, but the difficulty consists in ascertaining the actual condition of the various plates, for, as has been instanced above, a plate may be in such a position that unless the boiler be removed from its setting its condition cannot be ascertained with any approach to certainty. While speaking of the damage resulting from endeavours to patch up boilers, we may mention that some eighteen years since the boilers of a Thames steamer began leaking at the tube plates; the engineer in charge had the ferrules tightened up by driving in a mandril, the result being that the tube plates were cracked, and for that season the boat had to go slow, and for the ensuing season it was found necessary to fit new boilers in. In riveting boiler plates we would here strongly deprecate the practice of driving the "snaps" until they cut deeply into, and so weaken, the boiler plates.

Altérations to valve gear, where practicable, also form a favourite mode of improvement; but how far we can proceed in this direction will depend upon the arrangement of the existing parts, or whether it is thought worth while going to the expense of a new cylinder.

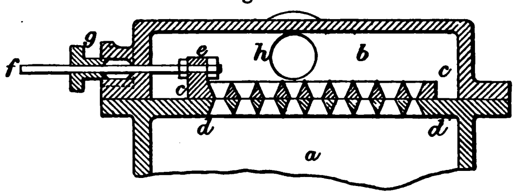
It may be advisable here to explain the direction to take in improving the valve gearing of an old engine.

Suppose the engine has been running with a short slide valve having "lap" on it to give a cut-off at, say, three-quarters of the stroke, and now we want to cut off at a fourth of the stroke, the best plan will be to take off all the "lap" from the slide valve, and put upon the back of the valve chest a separate expansion valve, which may be worked by cams or by an eccentric and link so as to furnish an adjustable cut-off. Expansion valves of this description are usually made in the form of a gridiron, and

therefore are called gridiron valves; the advantage attending their use is that a large steam area is opened or closed by a very small movement or travel of the valve, in a direction at right angles to the length of the slits. At Fig. 54 is shown a longitudinal section of a gridiron valve and its seating on the top of the ordinary valve-box.

a is part of the interior (broken off) of the ordinary valve chest, the back of which is pierced at *d d* with a number of narrow transverse slits as shown; *b* is the interior of the expansion valve chest, wherein is placed the

Fig. 54.



gridiron expansion valve *c c*; of course the bottom face of the expansion valve and the top of the ordinary valve chest are carefully surfaced so as to fit steam-tight. It will be seen that we show the slits as widening out in each direction from the surfaces of contact; this is to allow as free a passage as possible for the steam. *e* is a lug on the front end of the valve through which the valve-rod *f* passes, being thereto secured by nuts on each side of the lug as shown. The valve-rod passes through a stuffing-box *g* in the ordinary manner; *h* is the end of the steam pipe. An inspection of this section shows that by moving the valve the distance of *one* slit, the area of *nine* slits is opened or closed, practically, instantaneously.

The total area of the expansion valve should exceed that of the steam port; about one and a half times the area is a good proportion if there is room to obtain it.

We far prefer this description of cut-off valve to any of the noisy and fanciful trigger valves that have attracted the notice of some hyper-theoretical mechanicians, and we believe they are practically quite as efficient and much more durable. If we are called upon to deal with an engine having the top and bottom ports quite distinct from each other we can, of course, put in what valves we please, and manage the cut-off without a separate expansion valve. For particular forms of slide valves the reader is referred to the author's treatise on "Mechanical Engineering." *

It will be remembered that the rule for the thickness of steam-engine cylinders includes a considerable allowance for cross strain and vibration; let us now see what will be the difference of thickness necessary in a 25-inch cylinder between the pressures of 20 lbs. per square inch and 60 lbs. per square inch, following the rule given at page 80.

20 lbs. pressure of steam
25 inches diameter of cylinder

100
40

Constant 440)500(1·13 (a)
440

600
440

1600
1320

280
...

* "A Practical Treatise on Mechanical Engineering." By Francis Campin, C.E. Crosby Lockwood & Co., 7, Stationers' Hall Court, London.

```
   1·13      (a)
   5·00 square root of diameter
  ─────
 8)6·13
  ─────
    ·766 inches thickness
    ════

     60 lbs. pressure of steam
     25 inches diameter of cylinder
     ──
    300
   120
   ────
Constant 440)1500(3·40      (b)
             1320
             ────
              1800
              1760
              ────
                400
                ...
                ═══

   3·40      (b)
   5·00 square root of diameter.
  ─────
 8)8·40
  ─────
   1·05
   ════
```

The difference here is about ¼ inch, and it will often happen that in the first instance a good allowance (especially in old engines) has been made; at all events we can ascertain what *is* a safe pressure for the given cylinder from the following:—

RULE.—*From eight times the thickness in inches of the cylinder subtract the square root of the diameter in inches,*

divide the remainder by the diameter, and multiply the quotient by 440; the product will be the maximum safe-working pressure in lbs. per square inch.

Let the cylinder be 18 inches in diameter and ⅞ (·875) inch in thickness.

```
                   ·875 inch thickness of cylinder
                      8 constant
                  ─────
                  7·000
                  4·242 square root of diameter (18 ins.)
                  ─────
Diameter in ins. 18)2·758(·153*
                  18
                  ───
                   95
                   90
                   ───
                    58
                    54
                    ──
                     4
                    ••
                    ══

        ·153 * quotient
         440 constant
        ─────
        6120
       612
       ──────
       67·320 lbs. per square inch safe pressure.
       ══════
```

In making additions to machinery it will often be necessary to renew some of the already existing parts, as in the example of the expansion valve, where the ordinary

valve box must be altered to adapt it to receive the expansion valve chest.

In arranging, in the first place, the plant of a factory, thought should be given to the future probable extension of the works, for by so doing when the time for extension arrives much inconvenience and expense may be saved; for instance, if it is intended at some future time to couple another engine on to the first the bed plate should be cast with proper arrangements for the necessary connections, and the crank shaft should in the first instance be made sufficiently long and also of proper diameter to transmit the power of the *two* engines.

In some cases, especially engines for draining mines, the additional power is obtained by altering the rate of expansion as the amount of drainage water increases. Thus by putting down in the first instance an engine that will do the work with a cut-off of one-sixth or less, and as the work upon it increases altering that cut-off gradually up to, say a fourth or a third, the requisite additional power is obtained, though, of course, at the expense of economy, the consideration of which will vary in importance with the district in which the engine is working; thus at the mouth of a coal-pit, where slack costs next to nothing, economy of working is a very minor consideration.

In renewing the bearings of these *outdoor* engines, cast iron may be used instead of brass or gun-metal, and with advantage, for while it answers the purpose very well it is not so likely to be stolen. It may seem curious to talk about stealing engine bearings, but such cases have come under our own personal knowledge.

Some strange mistakes are made at times in small works where the machinery may be in the charge of an *intelligent* engine-driver, who thinks he is competent to

advise on the question of improvements. An amusing case of this sort occurred some years since at some water works near Dorking, in Surrey. The pumps were in the first instance driven by a water-wheel, and when it was considered that some auxiliary power was necessary, the proprietors put down *in the wheel pit* a small oscillating engine, about 12 inches in diameter, geared to the wheel shaft to help in turning it; the steam pipe, moreover, from the boiler to the cylinder was about 200 feet long. The absurdity of this arrangement is obvious, for, placed as that small oscillating cylinder was in the damp wheel pit, it would make a good surface condenser to condense such steam as reached it through the great length of boiler pipe.

The probability, if not indeed the fact, of the case is that the addition of the engine had the effect of increasing instead of diminishing the duty of the water-wheel.

CONCLUSION.

In concluding the present work it is necessary briefly to revert to the particular subjects brought within its scope, and also the general mode of dealing with them.

Our pages being devoted to the consideration of mechanical details, must necessarily appear in a somewhat piecemeal form, being classed according to the duties they are designed to perform, and not according to the machines of which they form component parts; thus a crank has the same duties to perform whether it be on a pump or a steam engine, and analogous remarks will apply to other mechanical details.

The larger and more costly works on steam and other machinery supply general ideas and illustrate examples of the work complete, but do not devote much space to the consideration of these minor details, which, although they appear insignificant in comparison with the whole bulk of the machine, are yet quite as essential to it as the apparently more important and active elements, for the failure of one bolt or one lug may cause the destruction of the entire machine.

The requirements of junior draughtsmen and foremen engineers have been especially considered in the preparation of this work, and every effort made to render all rules and calculations in the simplest and clearest forms.

In the space at our disposal it would evidently be out of the question to attempt to treat all kinds of mechanical details, but we have endeavoured to select such typical examples as may generally cover the ground contemplated.

A young draughtsman who has received only general instructions as to the preparation of working drawings for certain machinery, may often find himself at a loss, or in doubt, as to some of the details, and those who have not a lengthened experience will be unable to decide at sight some of the important though minor points that present themselves. In such a case this treatise should be of value either to instruct or to confirm an already formed opinion.

It is also highly desirable that apprentices, pupils, and students should inquire very carefully into the duties of all the details of the machines to which they have access, and from these to build up the complete work.

In inventing new machinery or adapting an existing class of machines to a new purpose, there is generally a good deal of tentative work, involving much alteration and a generally step-by-step mode of procedure, and more especially in new machines for manipulating materials, the properties of which have not previously been exhaustively ascertained. Thus, for instance, in certain flax machinery consisting of successive pairs of rollers, it is necessary for the rollers, which are variously grooved, to increase from pair to pair from the feed end towards the delivery, but it had to be determined by experiment what amount of increase should be given to the perimeters of each succeeding pair of rollers, for with too little the machine would be constantly choking up, and with too much the fibre of the material under treatment would be torn.

No practical man need be ashamed to have to make such trials, and even in such work as setting out complicated cams for new machines, it is convenient to make test models of thin wood or stout cardboard to *see* that they produce the required motion before putting the working drawings into the hands of the pattern makers and others engaged in the actual execution of the work.

The importance of *calculating* all elements that admit of being so treated can hardly be overrated, for it certainly is not satisfactory to *slavishly* copy the work of another man's hand without ascertaining that a proper margin of strength has been allowed without introducing an extravagant amount of material.

We cannot close our observations without endeavouring to impress upon all connected with engineering work the moral duty of acting honestly and conscientiously both as to materials and workmanship, for in the long run it will be found that trickery only leads to disgrace, whereas superior work cannot fail to establish a lasting reputation for the firms producing it. And the desire to produce the best quality should rest not only with the heads of firms, but also permeate all the assistants and workmen in their employ. Of course, these principles must emanate from the fountain-head, otherwise good materials will not be available, and to work upon bad stuff must prove discouraging to a skilled artisan.

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